INDUSTRIAL GAS TECHNOLOGY

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INTRODUCTION

B.C. Hydro organized with support from Energy, Mines & Resources Canada a conference & exhibition for large volume natural gas users. The theme of the conference was "Increasing Profits Through Innovation" and the objective was to inform industry in Western Canada and the Pacific Northwest of the most up-to-date gas utilization technology. This document includes all the papers presented at this conference.

The speakers program consisted of 27 presentations divided into the following 6 sessions: Plenary, Hot Water & Steam, Forest & Other Resource Based Products Applications, Transportation Applications, Cogeneration, and High Temperature. This program was designed to provide information on the latest equipment available to the complete spectrum of industry that exists in Western Canada and the Pacific Northwest such as: pulp & paper, lumber, food processing, metal processing, mining, agriculture, and transportation. The speakers came from the gas equipment industry, gas utilities, consultants, and research institutions.

Forty-two equipment suppliers and service organizations participated in the trade show portion of the conference. The following is a sample of the products that were exhibited and demonstrated: high efficiency burners, combustion control equipment, metering, energy management equipment, boilers, cogeneration equipment, heat exchangers, water heaters, NGV conversion equipment, NGV home compressors, and infrared heaters. A complete list of trade show participants is given in Appendix I.

The conference delegates came from a wide variety of industrial & institutional establishments. The majority were from British Columbia but there was representation from Ontario, Manitoba, Saskatchewan, Alberta, Washington, Oregon, and California. Most of the delegates were management and engineering personnel who are involved in the design and decision making process for plant process equipment. A complete list of delegates is given in Appendix II.

The following gas utilities and organizations supported the conference by supplying speakers and session chairmen and promoting the conference in their areas:

American Gas Association
B. C. Hydro
Canadian Gas Association
Canadian Western Natural Gas
Cascade Natural Gas Limited
Consumers Gas
Energy Mines & Resources
Forintek Canada Corp.
Gaz Metropolitain
Greater Winnipeg Gas Co. Ltd.
Inland Natural Gas Co. Ltd.
Institute of Gas Technology
Northwestern Utilities Ltd.
Pacific Gas & Electric
Pacific Northern Gas Ltd.
Saskatchewan Power Corp.
Union Gas Limited
Washington Natural Gas Co.

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STATUS OF INDUSTRIAL GAS DEVELOPMENT - USA

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by John B. O'Sullivan
Institute of Gas Technology

9/5/666

ABOUT THE SPEAKER:

John B. O'Sullivan graduated from Notre Dame University with a B.S. Degree, and from Iowa State University with M.S. and Ph.D Degrees.

Dr. O'Sullivan's early career was with the Research Division of Allis-Chalmers Manufacturing Company, Milwaukee, Wisconsin where he held positions ranging from Process Research Engineer to Manager, Mineral Processing Group.

From 1969 to 1980 he worked with the U.S. Army as a chemical engineer, then as Chief, Electrochemical Division, U.S. Army MERADCOM. He later became Chief, Petroleum and Environmental Technology Division, U.S. Army Mobility Equipment Research and Development Command, Fort Belvoir, Virginia.

Dr. O'Sullivan's work with the military included R&D on fuel cells and their application to vehicle propulsion systems, and water purification and fuel-handling equipment for the field army.

Dr. O'Sullivan joined the Institute of Gas Technology, Chicago in 1980 as Director, Energy Conversion and Storage Research where he administered IGT's research and development programs in fuel cells, energy conversion and storage, cogeneration systems and integral systems and economic analysis. Then, as Director, Advanced Technology Development, he was responsible for program development with industry and the government across most of IGT's technical areas.

Dr. O'Sullivan is now Assistant, Vice President, Energy Utilization Research and is responsible for administrative direction of research in the areas of combustion, energy storage, fuel cells, electrochemistry, efficient industrial and residential energy utilization, appliance research, space conditioning and indoor air quality, and natural gas fueled vehicles.

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Greetings Ladies and Gentleman - I am very pleased to have been invited here to make this presentation to you. Vancouver is also among my favorite cities and any excuse to visit here is a welcome one.

As should be obvious from the topic - a broad view of the status of R&D in the U.S. will prevent me from focusing on the fine details of any specific developmental program. That will be left to the speakers in the five sessions that are spaced over the next day and a half.

I should also like to try a somewhat different introductory approach. To understand the nature of the U.S. R&D effort one must place it in the proper perspective. First lets look at the consumption of energy in the Industrial Sector (Fig. 1). As you can see, this is a major sector of our energy use, it is also an area where the possibility of fuel switching to alternatives looms largest. Thus, the challenge to R&D to maintain gas markets is strong. The projected distribution of future fuel use (Fig. 2) indicates a clear dependence on gas. Natural gas is a domestic fuel; 95% of U.S. use is produced in the U.S. and 98% in North America. It will be an important fuel in the U.S. and Canada for many years in the future. An interesting way to view the relative resources of the U.S. and Canada is on a per-capita basis (Fig. 3) rather than a total volume of reserves. This abundance of natural gas in Canada should encourage the continuation of your own strong R&D program and the exploitation of successful technologic developments from other countries. Given the relative strategies of the U.S. government and the U.S. gas industry it is imperative to have a balanced program between research (long term) and development (short term) (Fig. 4). The gas industry is trying to achieve this - the budget at GRI (Fig. 5) is rationally balanced to address all the appropriate sectors. However, given the great expense of field demonstrations to validate the technical and economic viability of hardware development, the budget should be larger and growing. It has been effectively fixed the last few years as a result of the severe economic competition in the gas industry. The major responsible government agency is the U.S. Department of Energy (DOE). It has a budget of about $15 billion dollars; this covers many non energy efforts in support of the Department of Defense. Its energy R&D segment (Fig. 6) must support such areas as fusion, nuclear, high energy physics, etc. that have no direct beneficial impact on the gas industry. Even in the fossil energy and conservation areas major support is for coal and electricity as the preferred energy forms. As a result the U.S. Gas Industry lobbies hard for a reasonable allotment of gas related R&D (Fig. 7). As you can see the DOE usually requests little funding for direct gas related effort. This is primarily a result of government priorities - remember the current President has tried to abolish the DOE during his eight years in office. Fortunately the Congress continues to thwart the President's plans and provides fiscal support to maintain ongoing programs. This continual bickering and see-saw budgetary process does not provide an atmosphere conducive to innovative and effective research.

The major industry goals are shown in Fig. 8. A national goal is the third - to develop efficient processes that will assist us in recapturing lost or slipping markets, such as the production of steel. The major technical areas of interest to the gas industry are given in Fig. 9. I have chosen to include cogeneration and space conditioning since they span the areas of industrial and commercial uses and share the same needs in supporting hardware, namely inexpensive, reliable and efficient prime movers.

A balanced R&D program necessitates work on total systems, the individual system components when inadequate, and supporting efforts to permit the satisfactory solution of basic problems. In this presentation I will address most of these applications individually and in greater detail with specific developmental examples to provide emphasis. I will not cover natural gas vehicles, even though it is an area near and dear to me and where IGT has unusual expertise, because there is a whole session devoted to them.

An extremely important area of technology involves gas fired prime movers (Fig. 10). These are needed to make a large number of individual applications economic. Remember one must view the engine as a direct replacement for an electric motor. It must match or exceed the energy conversion efficiency of electric power generation and match the reliability of electric motors. This is much more readily accomplished at large sizes, but the problems increase in frequency and intensity at smaller sizes. Therefore, different approaches such as Stirling and rotary engines are called for at the residential/ commercial end of the spectrum. One can no longer, at smaller sizes, afford the ruggedness and thus cost of "industrial" engines and must instead substitute high production volume automotive derivative engines. These have been used in both cogeneration and chiller applications. The need for efficient and reliable military engines has supported the very expensive development of adiabatic diesels using ceramic components. The interest in higher efficiency turbines has extended the upper limit of inlet temperatures primarily with ceramic materials. Similarly the DOE has funded extensive automotive turbine development. These developments should eventually assist the gas industry in achieving economically some specific intermediate and large size applications.

Figure 11 lists a number of recent cogeneration projects supported by GRI and/or gas utilities. It is in these sizes that the initial cost/reliability issues surface most clearly. As you can see the whole range from innovative engines, to industrial, to automotive is covered. There is also emphasis on packaged units with a minimum of tailoring to the individual site. The only exception to this is the desire to provide in the design some flexibility in the power/heat ratio delivered. However, this is only attainable today in larger sizes, for example the GRI steam-injected turbine project. This unit delivers 3 MW at base load with all steam generated by waste heat going to the process. If more power is required process steam is diverted back to the turbine to increase output to 4 MW. The reduced process steam could be made up by supplementary boilers. An inexpensive bottoming cycle would permit the additional choice of increased power, or process steam, or absorption chilling in a cogeneration package.

Space conditioning development is of great interest to the gas industry in the U.S. especially those in the southern states who would be ecstatic with a summer load. Whereas the government's interest is with increasing efficiency of heating/cooling and with no special interest in whether the prime mover is an engine or an electric motor. Therefore, a full plate of options is being examined. DOE is pursuing the new and innovative cycles and fluids research and GRI the near term producible hardware. There is increasing gas industry interest in desiccant based systems. These desiccant systems have the advantage of providing humidity control and increased outdoor air exchange at no additional cost. With the concern for the tight house syndrome which results from implementing many energy saving procedures, this air exchange could be the incentive for R&D to reduce the high initial cost of some present desiccant designs. Another area in which there will be a burgeoning need is for new compressor design-fluid matching work as chlorofluorocarbons (CFC's) are banned or limited worldwide. DuPont has recently announced it will cease production of CFC's in the future.

Process development efforts are designed to encroach on technical areas previously the purview of the electric industry (or oil) or to limit the penetration of electricity into previously gas areas (Fig. 13). An example of the former is the use of oxy-gas burners to preheat/accelerate the melting of scrap in electric arc furnaces. The glass industry is a major user of gas, with some encroachment by electricity. GRI has put a major effort into glass preheating, reported on at the 1986 meeting sponsored by B.C. Hydro, and GRI is now funding R&D on an advanced glass melter. GRI has also supported research to examine the use of oxygen in glass melting to reduce NO_x emissions and increase efficiency. There are a broad range of investigations underway directed toward onsite oxygen production to compete with transport of liquid oxygen. Cryogenic plants have the economics of scale but are not cost effective at the small sizes needed by many industries. Separation technology, funded by many sources, based on pressure and/or temperature swing adsorption and various membrane processes has the promise of providing an economic competitor that will permit oxygen utilization by the process industry. Heat treating, which requires a controlled atmosphere, can be accomplished with recent developments in large radiant burners. Radiant burners for aluminum melting have been a materials challenge as a result of the chemically active atmosphere in aluminum furnaces. Immersion heaters are an even more severe challenge. Advances in paper drying will be covered in a later session; that industry along with food processing are two high volume areas of interest to the gas industry.

Burner developments (Fig. 13) are critical to successful gas fired applications. These efforts must cover not only the burner itself but the supporting hardware for heat recovery, chemical sensors, and combustion controls. The uniform flame characteristics of the fiber matrix burner have been successfully utilized in numerous applications. Pulse combustion has been similarly applied. Better heat recovery results in higher burner temperatures and increasingly severe materials requirements for the innovative designer. These temperature increases, and improved heat transfer to the load, are also accompanied by increased formation of nitrogen oxides (Fig. 14). Their formation is a direct function of time, temperature and oxygen-nitrogen concentrations. Careful control of the combustion process, through fuel and/or air staging and excess oxygen control, can minimize the formation of NO_x. In some processes whose very nature limits the degree of control, such as Municipal Solid Waste (MSW) plants, more innovative approaches must be taken. Use of reductants such as ammonia or urea in conjunction with catalysts is one approach. A selective catalyst that would work with methane as the reductant would be a boon to the gas industry. Another approach is reburn technology whereby a process is run near stoichiometric, with exhaust recirculation if necessary - gas is then injected as the reductant for NO_x. The excess gas is then combusted in an afterburner but at such a reduced temperature that reformation of NO_x is negligible. This approach is being tested in coal fired boilers coupled with lime injection for SO_x removal. It is also being examined at the pilot level in an MSW incinerator for NO_x reduction and final destruction of trace organics.

Incineration of industrial wastes will be a future growth area for natural gas utilization. As more and more landfills are closed, new ones will not open because of the NIMBY response (Not-In-My-Back-Yard). Therefore, many wastes, even those with low calorific value will be combusted with supplemental gas and the heat produced utilized elsewhere in the industrial plant. This will require the development of burner-heat recovery system designs that will permit the thorough combustion of difficult and often hazardous low-Btu wastes and the recovery of waste heat from chemically complex exhaust streams. Many of these wastes may produce low melting ash and/or readily condensible salt vapors that will dictate whether dry or slagging design configurations are most appropriate and will necessitate innovative design and control to minimize fouling of heat transfer surfaces.

This concludes my presentation. I thank you for your attention and I would be happy to address any questions you may have.

FIGURE 1.

ENERGY CONSUMPTION (QUADS):

	1980	1985	1990
RESIDENTIAL	9.335	9.207	9.269
COMMERCIAL	5.892	6.601	6.865
INDUSTRIAL	20.185	19.614	21.707
TRANSPORTATION	19.655	18.608	16.939

- 80% in Highway Fuels
- 85% of Highway Fuels in Gasoline.

FIGURE 2.

U.S. PRIMARY ENERGY CONSUMPTION

(QUADS)

	ACTUAL 1986	1990	2000	2010
PETROLEUM	32.3	32.0	35.3	38.3
GAS	16.5	17.1	18.5	18.9
COAL	17.3	17.4	20.2	25.4
NUCLEAR	4.5	6.4	6.4	6.1
RENEWABLES	6.4	6.8	8.3	9.7
TOTAL	77.0	79.7	88.7	98.4

1987 GRI BASELINE PROJECTION

FIGURE 3.

ENERGY SUPPLIES: PROVEN RESERVES

	QUADS/MILLION POPULATION		
	NATURAL GAS	CRUDE OIL	NATURAL GAS / CRUDE OIL
UNITED STATES	0.95	0.82	1.2
CANADA	3.95	1.64	2.4

FIGURE 4.

R&D STRATEGY

SHORT TERM *	LONG TERM
GAS RESEARCH INSTITUTE **	DEPARTMENT OF ENERGY
GAS UTILITIES	NATIONAL SCIENCE FOUNDATION

*FEDERAL ENERGY REGULATORY COMMISSION AND STATE PUBLIC UTILITIES COMMISSION DRIVEN

**ADVISORY BODIES HAVE SUPPORTED A 13 ± % LEVEL FOR BASIC RESEARCH

FIGURE 5.

GRI R&D BUDGET

	1988		1989[a]	
SUPPLY	$M	%	$M	%
END USE[b]	37.3	25.8	40.2	27.4
	82.2	56.8	80.5	54.9
OPERATIONS	18.7	12.9	20.0	13.6
CROSSCUTTING RESEARCH	6.5	4.5	6.0	4.1
TOTAL[c]	144.7	100.0	146.7	100.0

[a] PRELIMINARY — NOT FERC APPROVED

[b] INDUSTRIAL, ~ 30% OF SECTOR

[c] R&D INVESTMENT; ADMINISTRATION EXCLUDED

FIGURE 6.

DOE ENERGY R&D BUDGET

(IN MILLIONS)

	FY 1988 ESTIMATE	FY 1989 REQUEST
BASIC SCIENCE RESEARCH	1,367.2	1,713.3
FUSION, NUCLEAR AND OTHER	1,074.1	1,096.5
CLEAN COAL TECHNOLOGY	50.0*	525.0
FOSSIL ENERGY	354.1	168.0
CONSERVATION	156.2	86.3
RENEWABLE ENERGY	149.0	124.9
	3,150.6	3,714.0

* ADDITIONAL $146 MILLION — CARRYOVER FROM PRIOR APPROPRIATION

FIGURE 7.

DOE (GAS-RELATED) BUDGET

FY	CONGRESSIONAL APPROPRIATION 88	DEPARTMENT REQUEST 89	INDUSTRY TESTIMONY*
FOSSIL ENERGY	63	17	67
— FUEL CELLS	20.5	6.6	22.7
— ADVANCED RESEARCH AND TECHNOLOGY DEVELOPMENT	6	5.1	6
NATURAL GAS VEHICLES	0	0	5
CONSERVATION	20.5	7.5	24
— GAS HEAT PUMPS	6	2.7	7
— INDUSTRIAL COGENERATION	5	0.8	5
GEOPRESSURED	5	2.9	10.4
BIO FUELS	9.5	4.3	8
GAS-RELATED R&D	98	34	114.4

*A.G.A., GRI, INGAA, IGT

FIGURE 8.

GAS INDUSTRY GOALS

- RETAIN EXISTING MARKETS
- ENTER NEW MARKETS
- ENHANCE COMPETITION WITH FOREIGN TECHNOLOGY/ MANUFACTURING
- DEVISE PROCESSES AND/OR DEVICES THAT EXPLOIT UNIQUE CAPABILITIES OF GAS

FIGURE 9.

INDUSTRIAL TECHNOLOGY

APPLICATIONS	COMPONENTS	SUPPORTING
HEAT PUMPS/CHILLERS	PRIME MOVERS	MATERIALS
		— METALS
SPACE CONDITIONING	COMPRESSORS	— CERAMICS
	HEAT EXCHANGERS	CONTROLS
COGENERATION	FUEL CELLS	MODELING
		ABSORBENTS
PROCESS HEATING	BURNERS	
		ADSORBENTS
	RECUPERATORS/REGENERATORS	
	OXYGEN PRODUCTION	
EMISSIONS REDUCTION	CATALYSTS	
	REBURN TECHNOLOGY	
	COMBUSTION MODIFICATION	
NATURAL GAS VEHICLES	DEDICATED ENGINES	
	STORAGE SYSTEMS	

FIGURE 10.

PRIME MOVERS

- DIESEL
  - ADIABATIC
- GAS TURBINE
  - COMPACT, REGENERATIVE
- STIRLING
- SPARK IGNITION
  - RECIPROCATING
  - ROTARY

FOR

- COGENERATION
- HEAT PUMPS
- CHILLERS
- VARIABLE SPEED APPLICATIONS

FIGURE 11.

COGENERATION PROJECTS

ADVANCED TECOGEN 75 kW (CHEVROLET 454 in.^{3})

FIELD TEST 310 kW (CATERPILLAR ENGINE)

ECONOGEN 35 kW (PRESTRATIFIED CHARGE)

SMALL PACKAGE 30 kW (V-6 MARINE ENGINE)

TECOGEN PREPACKAGED 200 kW (CUMMINS ENGINE)

SOLAR 200 kW (GAS TURBINE)

AIR RESEARCH/GARRETT 40 kW (TURBO ALTERNATOR)

FIGURE 12.

SPACE CONDITIONING

VAPOR COMPRESSION
- AZEOTROPIC
- NON-AZEOTROPIC

ABSORPTION
- SINGLE
- DUAL CYCLE
- HYPERABSORPTION

DESICCANTS
- SOLIDS
- LIQUIDS

AND COMBINATIONS (OF THE ABOVE)

FIGURE 13.

PROCESS DEVELOPMENT

GLASS PREHEATING AND MELTING
FOOD AND PAPER DRYING
CEMENT
ALUMINUM MELTER
IRON/STEEL TREATING AND MELTING

FIGURE 14.

BURNERS

DESIGN	SUPPORTING
DIRECT	RECUPERATOR
— FIBER MATRIX	— CERAMIC
— PULSE COMBUSTION	— METALLIC
	— CORROSION RESISTANT
RADIANT	REGENERATOR
— CERAMIC	
— METALLIC	
GAS/OXYGEN ENRICHED	

FIGURE 15.

EMISSIONS CONTROL

GAS	OTHER FUELS
STAGED COMBUSTION	CO-FIRING
OXYGEN ENRICHMENT	REBURN
	— MSW
CATALYST	— COAL

SECTION

II

THE HI-TECH CONNECTION

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Colin Mason
B. C. Hydro

9/5/666

ABOUT THE SPEAKER:

Colin Mason was born in England and graduated from the University of Leeds with a Bachelor of Science Degree in Fuel and Metallurgy.

He came to Canada in 1953 and has been registered as a Professional Engineer since 1955.

He has worked with utilities, manufacturers and consulting firms in various engineering capacities, relating primarily to the use of natural gas for industrial heating and processing.

Colin joined B. C. Hydro in 1973 and is an Industrial Process Engineer in the Gas Supply and Marketing Division of B. C. Hydro Gas Operations.

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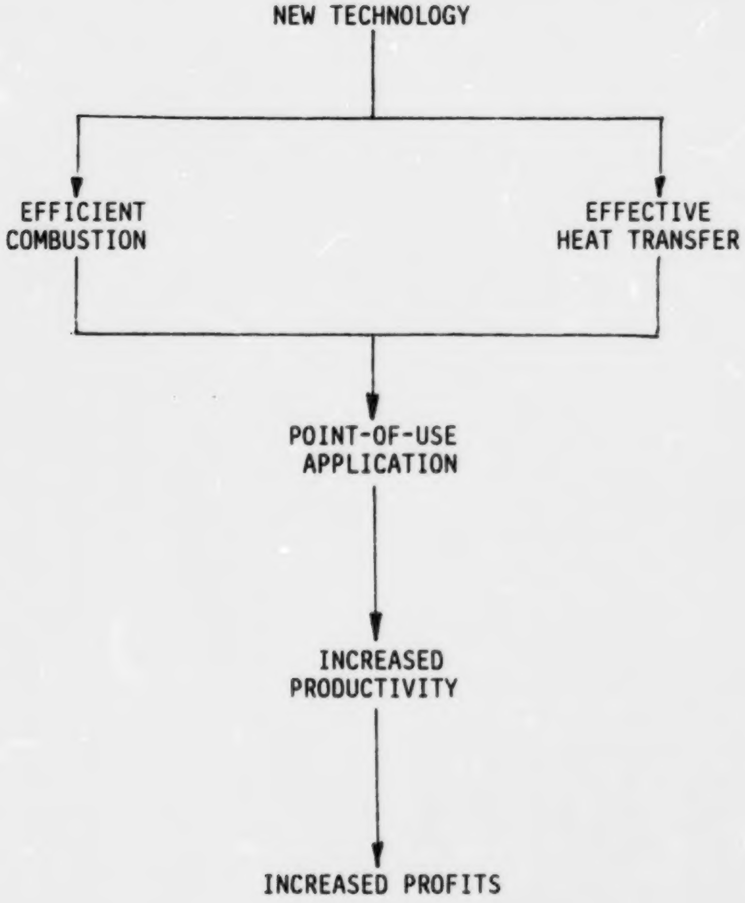
THE HI-TECH CONNECTION

INTRODUCTION

Over the next two days we will be discussing the ways in which new gas technology can be used to increase profits by increasing productivity.

The route from new technology to increased profits can be shown like this:


NEW TECHNOLOGY
EFFICIENT COMBUSTION
EFFECTIVE HEAT TRANSFER
POINT-OF-USE APPLICATION
INCREASED PRODUCTIVITY
INCREASED PROFITS

We see that new technology affects the two prime considerations of combustion and heat transfer. When these are optimized and can be applied to a point-of-use application, increased productivity and profits will result.

EFFICIENT COMBUSTION

For the complete combustion of natural gas we need sufficient oxygen (usually from atmospheric air, or, under certain circumstances, from oxygen enriched air) to ensure that the combustible elements in the gas, carbon and hydrogen, are converted to carbon dioxide and water vapour. (As oxygen-enriched combustion air will be discussed later in the seminar, the present discussion will be limited to combustion using atmospheric air only.)

We can show the combustion process for natural gas (essentially methane, CH_4) in the form of a chemical equation. For complete combustion in atmospheric air:

$$CH_4 + 2O_2 + 7.54N_2 = CO_2 + 2H_2O + 7.54N_2 + \text{Heat}$$

In practice, it is not usually possible to mix air and fuel at the exact (stoichiometric) ratio sufficiently well within the confines of a combustion chamber to assure complete combustion, and some excess air is required. The combustion equation may then be written:

$$\begin{aligned} &CH_4 + 2O_2 + 7.54N_2 + \text{Excess Air} \\ &\quad = CO_2 + H_2O + 7.54N_2 + \text{Excess Air} + \text{Heat} \end{aligned}$$

Again, in practice, it is not possible to achieve the completeness of combustion as shown in the above equation, and a complete analysis of the products of combustion will show the presence of carbon monoxide. Carbon monoxide is an intermediate product of the combustion process and is normally undesirable. Its presence in the flue gases represents unburned fuel which has to be paid for. Fortunately, however, we are able to take advantage of this characteristic of the combustion process and use carbon monoxide measurement and control for optimizing combustion efficiency. This technology will be discussed this afternoon by Bailey Controls.

A "rule of thumb" for gaseous fuels is to allow 10 cubic feet of combustion air for every 1000 Btu of fuel. (As natural gas has a nominal heating value of 1000 Btu/ft^3, the air requirement would be 10 ft^3 air per 1 ft^3 of gas).

The ratio between the heat absorbed by a system and the heat input to the system is the combustion efficiency. In the case of a boiler, the heat absorbed by a system would include heating the boiler water and radiation losses from the boiler surface.

Expressed in another way, we may say:

$$\text{Combustion Efficiency} = \frac{\text{gross heat input-heat loss in stack}}{\text{gross heat input}} \times 100\%$$

Gross heat input is the fuel input to burners. Heat is lost in the stack through:

(a) The heat content in the products of combustion plus excess combustion air supplied to the burner.

(b) Incomplete combustion. This results in carbon monoxide, hydrogen and, in some cases carbon particles, passing out of the exhaust.

For a given fuel, and assuming no incomplete combustion loss, the combustion efficiency can be obtained by taking accurate measurements of the oxygen (O_2) content, the flue gas temperature, and applying these data to a combustion efficiency chart for the fuel.

The three factors important to maximize combustion efficiency are:

(a) Excess air in the combustion chamber should be kept to a minimum.

(b) Stack gas temperature should be as low as possible.

(c) The amount of unburned gaseous combustibles (incomplete combustion) in the exit flue gas should be as low as practicable.

Local examples: Award Winners Aljane Greenhouses; B. C. Ministry of Agriculture; Davis Wire; Ravenek Greenhouses; Winvan Paving.

Oxides of nitrogen are an undesirable product of the combustion process and although these emissions do not materially contribute to combustion inefficiency, these should be kept below acceptable limits (Reference talks by Todd and John Zink).

It is evident from the methane/air combustion equation that if the volume of nitrogen can be decreased (i.e. the oxygen content of the air increased) the volume of exhaust gases - and associated heat loss - will decrease.

Oxygen enrichment of air causes an increase in flame temperature which, in turn, results in increased heat transfer and heating efficiency. The benefits of oxygen enrichment will be discussed tomorrow in the talk by Canadian Liquid Air.

EFFECTIVE HEAT TRANSFER

The energy transport resulting from a temperature gradient in a system is known as heat transfer. There are three modes of heat transfer: conduction, convective and radiation. In practice, convection and radiation are the major considerations.

Conduction is the transfer of heat through a substance without any appreciable displacement of any of its molecules.

$$q = \frac{k \times A \times \Delta T}{L}$$

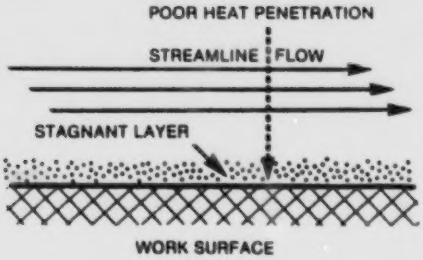
where: q = amount of heat conducted
k = thermal conductivity
A = area through which heat is conducted
ΔT = temperature difference
L = length of heat flow path

For maximum heat transfer by conduction it is evident that for a given material, the area and the temperature difference should be as large as possible.

Convection is heat transfer by moving and mixing masses of fluid and is a combination of conduction and fluid motion.

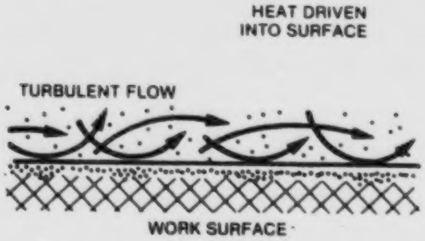
When the motion of the fluid (e.g. air or a liquid) is due to the natural buoyancy induced by temperature differentials of the heated fluids, the process is termed natural convection.

Forced convection occurs when fluid motion is assisted by mechanical means, such as a fan, and the amount of heat transferred will increase with the velocity of fluid movement.



(a)

Poor heat penetration caused by stagnant film layer.



(b)

Efficiency of heat transfer improved by increasing velocity of heating gases.

The main obstacle to efficient convective heat transfer is the thin film of gas on the surface to be heated. When the velocity of the heating gases is sufficiently high, however, the insulating film is reduced or removed. The efficiency is thus improved and a high temperature gradient between the furnace and the work is not required.

As a rough guide, for processes operating up to 600°C (1112°F), heat transfer by convection is of the greatest importance. As the temperature increases above 600°C, heat transfer by radiation becomes more significant.

The use of high velocity gas burners, which will be discussed by Bloom Engineering tomorrow, is one way in which convective heat transfer can be improved.

Radiation heat transfer is the transfer of heat by means of electromagnetic waves, which are similar to light waves. The quantity of heat available from radiation is proportional to the fourth power of the absolute temperature of the heat source, whereas conduction and convection are dependent on a temperature difference as a "driving force" for heat transfer.

Heat transferred by radiation:

$$Q = \text{radiation coefficient} \times A \times 10^{-8} (T_s^4 - T_r^4)$$

where: Q = amount of heat transferred by radiation
A = area of heat flow path
T_s = absolute temperature of source
T_r = absolute temperature of receiver

The radiation coefficient takes into account the emissivities of the materials employed and the relative positions of the radiating and receiving surfaces.

The equation clearly shows that the most effective heat transfer by radiation will be accomplished with the highest practicable temperature. (Reference talk by Krieger on infrared for paper drying.) Preheating combustion air raises flame temperature and accordingly increases the heat transfer rates. (This benefit is in addition to that of improving combustion efficiency by lowering stack temperature. Reference talk by North American on regenerative burners.)

POINT-OF-USE APPLICATION

A point-of-use application is the use of a direct-fired gas system for process heating rather than an indirect system.

Example of point-of-use applications are the replacement of steam by direct-fired gas systems for water heating, paper drying, etc. At the Awards Presentation we will be showing how Money's Mushrooms improved productivity significantly by replacing steam with direct gas-firing for mushroom sterilization.

The fuel-to-process efficiency is much higher with the point-of-use application. For example, in a water heating application we would be comparing an overall efficiency in the order of 50% for a system using steam with 80% or more for a direct gas system (i.e. fuel saving of over 35%). The talk by F. L. Energie will be illustrated with many examples related to liquid heating with direct gas-firing.

IN CONCLUSION

We have looked at these key elements to our program for increased profits using the latest technology. So now let us use the next two days to see the role which new gas technology plays in the real world of industrial gas utilization.

SECTION

III

PROGRESS IN INTERNATIONAL GAS TECHNOLOGY FOR INDUSTRIAL APPLICATIONS

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Sam Bernstein
Energy International Inc.

9/5/666

ABOUT THE SPEAKER:

Samuel Bernstein holds a Ph.D. in Fluid Mechanics from the University of Washington which he received in 1975. He also has MS and BS degrees in Aerospace Engineering from Iowa State University.

In the academic field, he has been an Instructor at Iowa State University and a Research Associate at the University of Washington. Dr. Bernstein joined Flow Industries Inc., of Kent, Washtington where he was Manager of the Energy Technology Department until 1981.

From 1981 to 1983 he was Director, Planning and Appraisal, Gas Research Institute, Chicago and since 1983 to the present time has been President of Energy International Inc. of Bellevue, Washington.

Dr. Bernstein's specialties include fluid mechanics, multi-phase flows, combustion, heat and mass transfer and efficient electric and gas utilization.

He is the co-holder of a U.S. Patent "Cogeneration Micrwave Food Dryer".

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PROGRESS IN INTERNATIONAL GAS TECHNOLOGY FOR INDUSTRIAL APPLICATIONS

Presented At:

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INTERNATIONAL INDUSTRIAL GAS TECHNOLOGY DEVELOPMENTS

ABSTRACT

The industrial gas market is experiencing keen competition worldwide from other fuel-sources. In response, significant advances in the field of industrial gas technology are being performed by equipment developers and gas utilities to improve the energy value of the gas based equipment. In this paper, technology development trends and application examples in the area of industrial gas equipment are presented. The discussion focuses on recent development philosophy and trends in Japan as a case study to illustrate how end user demand, competition between fuel sources, and regulatory requirements create an environment to foster advanced developments for specific market niches. The presentation also includes specific examples of burner technology for boilers and furnaces. A synopsis of advanced burners for enhanced heat transfer and low NO_x emission, and examples of their applications to industrial thermal processes are given to illustrate how the international experience may influence the available choices for the North American industrial gas equipment.

1. INTRODUCTION

For advanced gas equipment to be accepted by the industrial user, the fundamental design approach of the equipment must be responsive to a number of key market needs as follows:

1. Productivity enhancements
2. Product quality improvements for specific applications
3. Operating cost savings
4. Low initial cost requirements
5. Compactness (smaller footprints)
6. Low emission and other environmental impacts

The Japanese gas industry, as well as the gas industry in a number of other countries, has recognized the importance of diversifying and upgrading the equipment technology base of their customers and of becoming more market responsive. This is particularly true at a time of fierce inter-fuel competition and/or rising energy prices.

Because factory floor space, costs, and energy prices are much greater in Japan compared to North America, a detailed analysis of the offshore technology trends in Japan, provides a potential indicator of future trends in North America. The analysis provides a valuable case study on advanced equipment availability and application experiences which may influence the design philosophy in the North American market in the future.

Because of the connection between energy costs and policies, this presentation consists of five parts: (1) Energy Utilization and Pricing Trends in Japan, (2) Japanese National Energy Policy, (3) Analysis of Industrial Market Needs, (4) Offshore Equipment Development Examples, and (5) Future Trends.

The objective of this presentation is to describe the development of advanced gas technology from Japan and to assess how the offshore experience may influence the future's available equipment portfolio in the North American industrial gas market. Detailed analysis of differences and similarities between the market needs, energy prices, equipment design approaches, and application experiences is included to illustrate the possible ways which the state-of-the-art, offshore industrial gas technology may influence future applications in the North American market.

2. ENERGY UTILIZATION AND PRICING TRENDS

Due to its obvious impact on the industrial gas marketing strategy, an overview of the changing energy situation in Japan will must be cited first. Although the sense of urgency over energy issues has subsided somewhat in Japan, interfuel competition continues to be strong. Today's Japanese gas industry is strongly based on LNG imports. LNG has been imported to Japan since 1969 in an effort to convert from low Btu manufactured gas. To date, approximately 80% of the town gas market has been converted to LNG. The motivation behind the LNG conversion was to promote: (1) the diversification from a foreign oil-dependent energy supply system, (2) the favorable environment impact of LNG utilization compared to production and burning of manufactured gas, (3) a reliable and long term supply contract of LNG with key suppliers, and (4) the higher heating value available from LNG for a greater energy supply capacity with the existing underground distribution system.

Aided by the concerted effort to promote the use of LNG across all sectors, gas sales since 1975 has grown at a brisk rate. Figure 1 illustrates the growth of gas sales compared to the growth of real Gross National Product (GNP) and the primary energy supply. Since 1973, the first oil crisis in Japan, a significant effort has been expended on energy conservation measures across the board. As can be observed from the Figure, the primary energy supply leveled off while real GNP continued to grow at approximately 5-7% annualized rate. This indicates that, in general, the Japanese economy became more productive (per Btu of energy consumed) since the first oil crisis. Of course, the same trend of energy use is well documented in the United States and most of all other industrialized nations. The significant difference in the energy supply situation in Japan, however, is that the growth of gas sales outpaced the growth in real GNP. This is due to the Japanese central government's effort to diversify from oil-dependent economy to a more balanced and reliable energy supply infrastructure.

A comparison of the primary energy use breakdown between the United States and Japan is given in Figure 2. Japan's total energy consumption in 1985 was approximately 17 Quads (about 1/4 of the U.S. annual consumption). Noteworthy differences are: (1) Japan is still dependent on imported oil for nearly 60% of its energy source compared to slightly over 40% for the U.S. and (2) natural gas supply in Japan is still a relatively small market as a primary energy source.

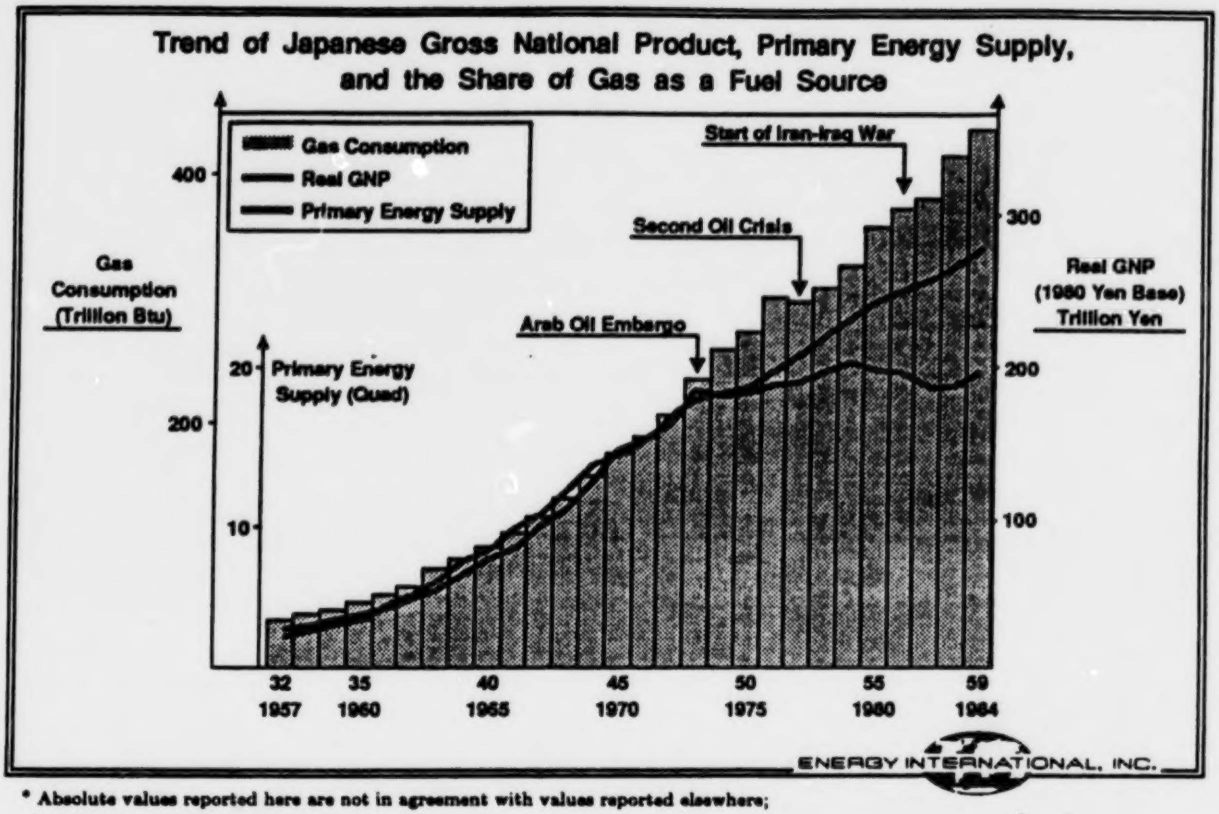
The difference between the U.S. and Japan in terms of gas utilization areas is more prominent as illustrated in Figure 3. The largest gas utilization sector in the United States is the commercial/residential applications which consume nearly 50% of all natural gas sold whereas the largest gas user in Japan is the electric utilities who consume nearly 3/4 of all gas used.

For reasons of a reliable and long-term energy supply, the Japanese government continues to promote gas utilization. In fact, there is a tremendous opportunity for Japanese gas utilities to market LNG sales to the industrial customers. Because of its reliable long term supply picture and stable prices, gas is gaining momentum in expanding sales to the industrial sector. As shown in Figure 4, the Japanese central government projects the following trends:

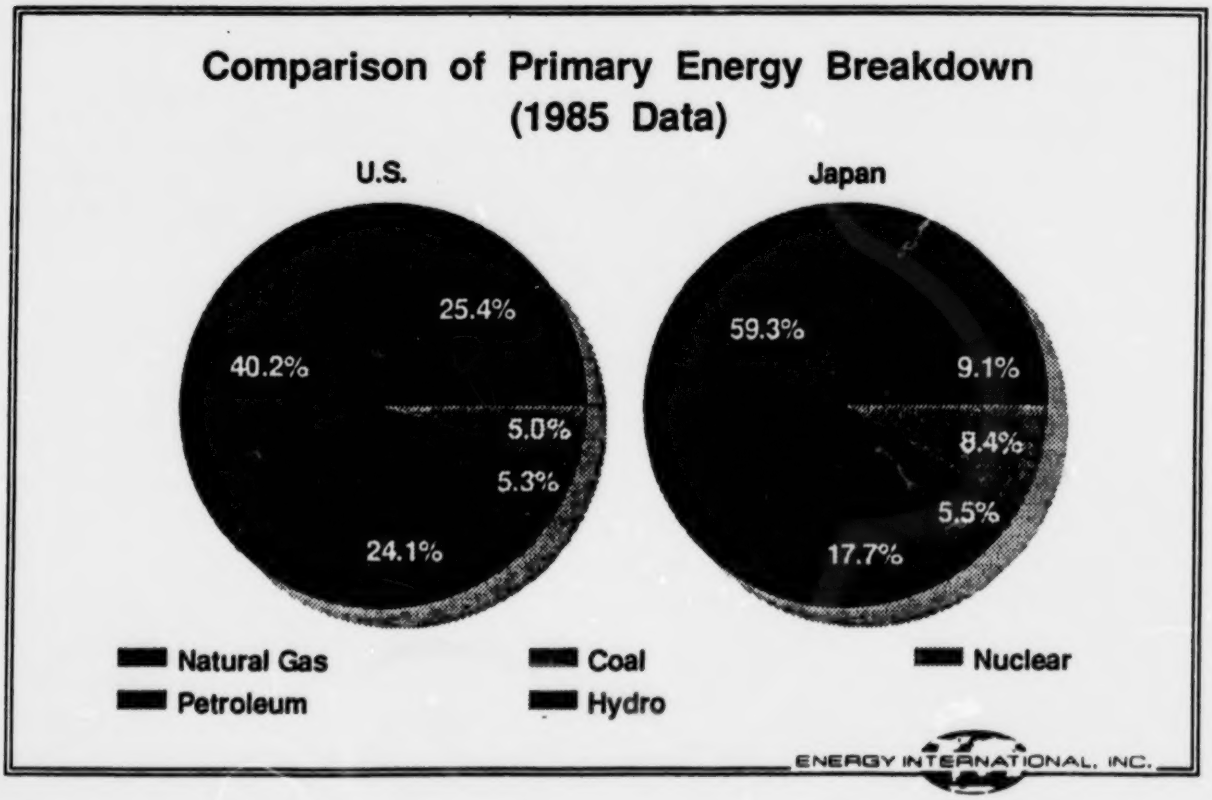
- A steady increase of gas utilization into year 2,000 (9.2 % of total energy consumption in 1984 vs. projected 11% by the year 2000).
- Oil imports will decrease from 59.6% of total energy consumption in 1984 to 46% by the year 2000).
- Nuclear power generation capacity will increase from 11.3% of total energy consumption to 16% by the year 2000.

Consequently, the greatest marketing opportunity for the Japanese gas utilities is in the area of fuel switching from oil to LNG, particularly in the industrial sector. Furthermore, the greatest area of competition is in the projected surplus of nuclear power during off-peak hours which can be sold to industrial customers at extremely attractive prices. It is in this background that the Japanese industrial gas technology development is being developed and promoted. The current advances in gas technology appear to be focused on:

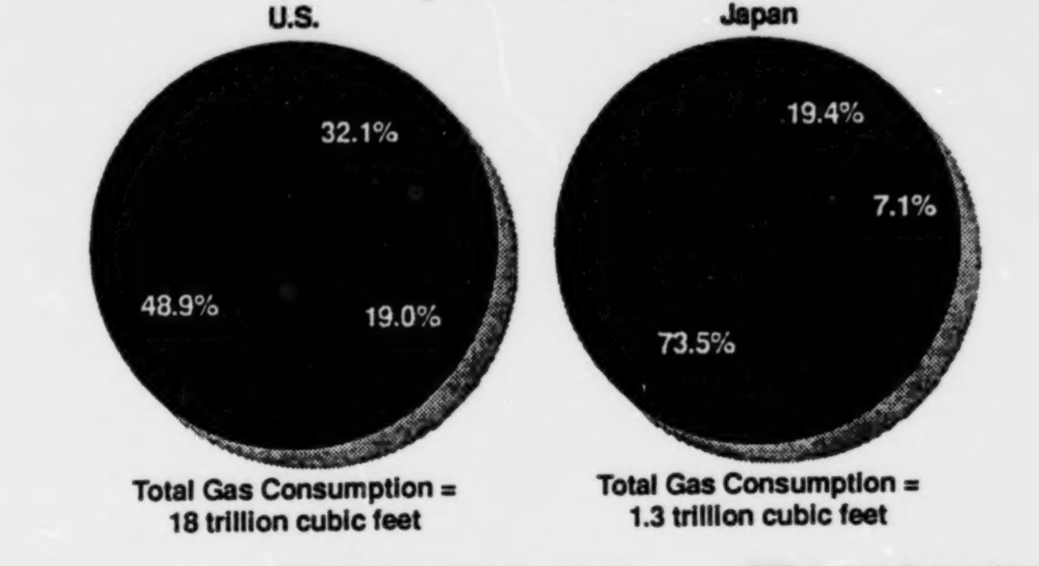
- Advanced gas burner development to effectively replace oil-fired boilers and furnaces.


Trend of Japanese Gross National Product, Primary Energy Supply, and the Share of Gas as a Fuel Source

* Absolute values reported here are not in agreement with values reported elsewhere; however, the trends are in agreement.


Comparison of Primary Energy Breakdown (1985 Data)

Comparison of Gas Utilization (1984 Data)



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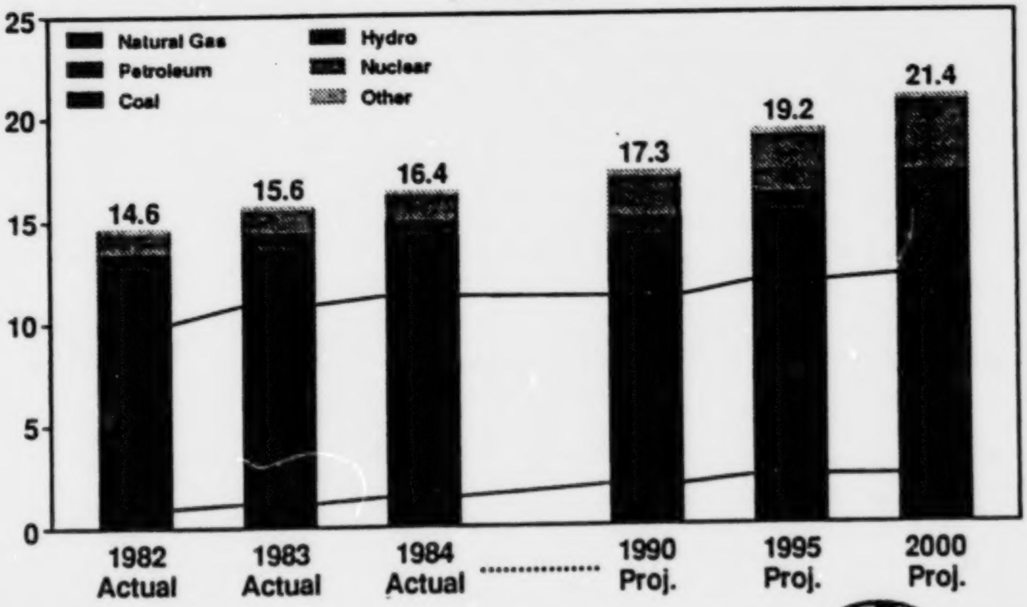
Projection of Primary Energy Supply in Japan

(In Quads)



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- Innovative gas cooling technology packages to expand the use of LNG for commercial and residential cooling customers to compete against the relatively expensive electric cooling systems.

- Emphasize the benefits of gas use in many different types of industrial applications in anticipation of aggressive off-peak electricity sales.

Because of its lack of domestic energy resources, Japanese energy prices are comparatively high. Figure 5 shows the range of energy prices available to industrial customers in Japan. Large energy users, such as the industrial customers, are allowed to negotiate independent purchase contracts with local utilities. Therefore, there is a wide range of price differential for each fuel source, which is a function of seasonal and daily peak usage, annual energy consumption level, local competition between various energy supply systems, etc. With today's low oil prices aided further by the high value of the Yen, Japanese industrial customers are enjoying relative price relief for their energy sources. Because some LNG purchase contracts between the Japanese utilities and foreign suppliers are tied to the price of crude oil, LNG prices are also declining. Even then, the lowest LNG price is over 6 dollars (U.S.) per million Btu. The highest industrial LNG rate is over 11 dollars (U.S.) per million Btu, which is matched by the cheap off-peak electricity surplus at 12 dollars (U.S.) on the million Btu basis. Average industrial electric rate is approximately 8 cents (U.S.) per kilowatt-hour. The trend of cheap electricity during off-peak hours is thus creating significant competition against LNG.

3. JAPANESE NATIONAL ENERGY POLICY

Another parameter which has influenced the energy supply structure in Japan is that of the national energy policy. Having learned a costly lesson during the 1970s from two major oil crises, the Japanese central government has established a national policy of diversification from an oil-dependent economy. This policy was issued as a Cabinet Decree in 1981 which allowed the Ministry of International Trade and Industry (MITI) to implement a revised utility rate schedule, tax incentives, and financial assistance to promote the use of LNG and nuclear base load. The major incentives which resulted include:

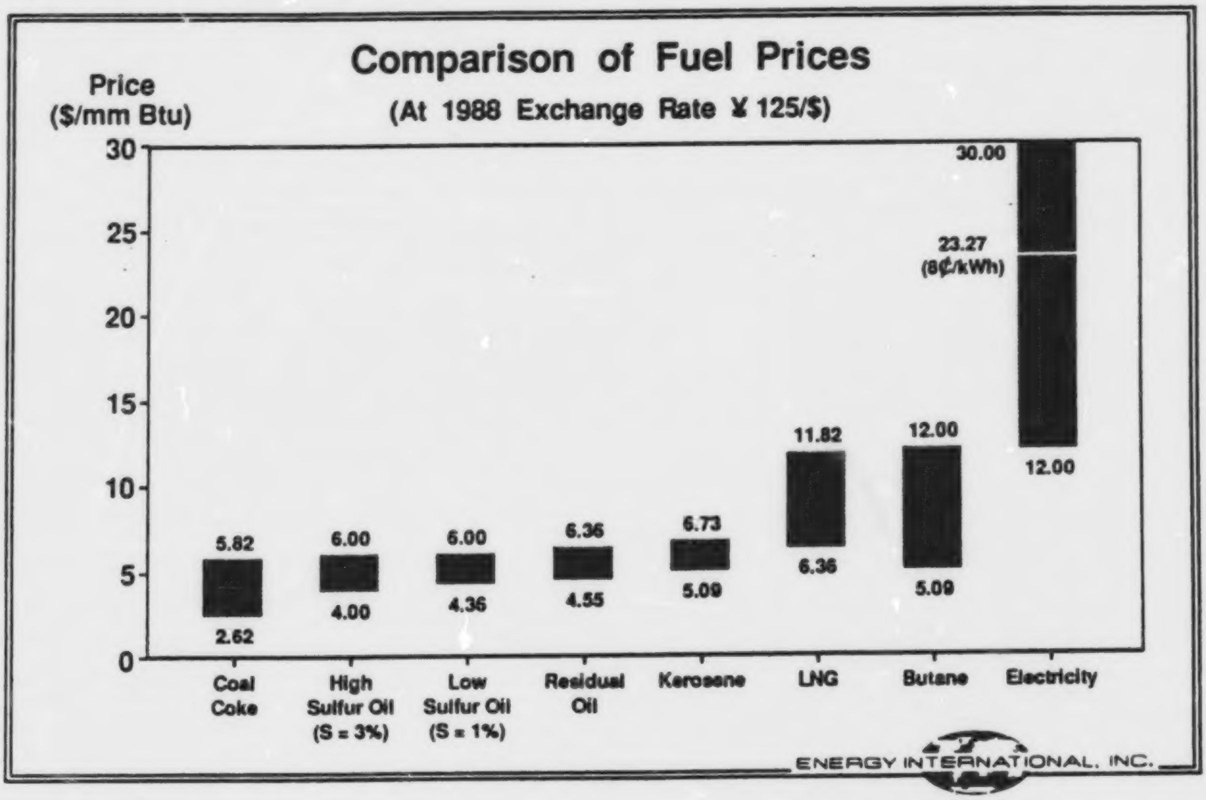
- Favorable gas rates and the elimination of national energy sales tax for gas used for cooling.

- Establishment of three basic daily rate periods for electricity: peak rate, off-peak rate, and off-off-peak rate.

- Assistance through government-subsidized banking agencies to provide low-interest loans for the purchase of gas cooling equipment and energy efficient gas boilers and furnaces.

As a result, gas utilization in the commercial sector for building cooling has grown substantially, surplus nuclear power is promoted as a cheap off-peak energy source, and seasonal load leveling is continuing for both gas and electricity.

Because of these changes in the energy supply structure and the government's policy to control the energy use mix, the Japanese industrial market is characterized as an artificial environment in which advanced gas technology can evolve. While the market growth of LNG utilization has been impressive, whether the same rate of growth could have been achieved without the aid of incentives is an open question.



4. ANALYSIS OF INDUSTRIAL MARKET NEEDS

The advanced gas equipment must perform more economically than the electric equipment under the prevailing off-peak electric rates. It must also perform more economically than the oil fired equipment to take advantage of market opportunity. In order to meet the challenge, the focus of gas technology development in Japan is placed on energy conservation, productivity enhancement, and product quality improvement.

By comparison in the North American market, the energy cost constitutes a relatively small portion of the manufacturing cost of goods and services. Thus, typical industrial equipment for basic manufacturing is marketed on the basis of productivity enhancement and product quality improvement, rather than energy efficiency. However, in Japan, the average industrial fuel rates are approximately three times the U.S. average and, consequently, energy efficiency is often the key to successful marketing.

In reviewing recent developments in Japanese gas technology for industrial applications and how they are being marketed, one can quickly reach a conclusion that industrial equipment design needs are prioritized in the following order.

1. Operating cost savings
2. Productivity enhancement
3. Product quality improvement
4. Initial cost
5. Compactness
6. Low emission and other environmental impacts

The priority listing reflects the industrial economics of Japan, i.e., expensive energy, attention to high productivity and quality of goods, and expensive real estate for factory construction.

In summary, energy cost is still a significant part of manufacturing cost in Japan and equipment is promoted by promising operating cost savings. Furthermore, for the gas industry at large, today's industrial technology development must address application-specific economics to be successful.

5. JAPANESE EQUIPMENT DEVELOPMENT EXAMPLES

In this section, examples of Japanese gas technology applications in industrial thermal processing sector are presented. To summarize the complete perspective on the significant developments by the gas industry, it is noted that:

1. In the residential market, many innovative gas space conditioning and water heating appliances have been developed. Examples include residential gas engine heat pumps, condensing type instantaneous water heaters, ultra-compact high capacity burners, catalytic burners for room heating, and radiant plate space heaters for infrared heating.

2. In the commercial market, the emphasis is placed on the cooling load applications. Examples include commercial size gas engine heat pumps, gas absorption chillers, absorption heat pumps, fuel cells, cogeneration district heating and cooling systems, and pulse combustion water heaters.

3. In the industrial market, the focus of the development has been one wide range of advanced burner systems and associated equipment.

The next section discusses sample developments and application studies in the area of burner technology.

5.1 Industrial Burner Technology Summary

A summary of industrial burner technology is shown in Figure 6. The wide variety of burners available from Japan and elsewhere can be grouped into three categories which are distinguished by the predominant heat transfer mechanism. They are: (1) convective heating group, (2) radiative heating group, and (3) hybrid heating group. Figure 6 illustrates such grouping of the state-of-the-art burners by mapping the convective heat transfer effectiveness on the left vertical axis, radiative heat transfer effectiveness on the right vertical axis, and corresponding flame temperatures on the horizontal axis.

The resultant mapping shows that:

- Of the convective heating group, a number of low NO_x burners have evolved (particularly for boilers for which NO_x standards can be quite stringent) using techniques to lower the peak flame temperature. Example techniques for NO_x control include exhaust gas recirculation, two-stage combustion, sub-stoichiometric combustion, and lean combustion.

- The innovative burner technology which accomplishes high convective heating capability and NO_x control is the pulse combustion burners. While this burner technology is not unique to the Japanese development, a large number of burner applications in 1-10 million Btu/hr size exist in Japan. These burners are becoming increasingly popular in water heating, batch preheating, and drying in the food processing and textile industries.

- For applications which require high temperature processing, such as in metal heating and treatment furnaces, high convective burners and high temperature burners are gaining acceptance.

- For efficient radiative heating effectiveness, catalytic combustion equipment is used as a large space heater. Because these burners take advantage of catalytic reaction of methane, rather than combustion, flame temperature is low, reaction is nearly 100%, and dominant heating is due to infrared (and far infrared) emission.

- For applications which require non-oxidizing treatment or heating temperature uniformity, a number of infrared burners are available, such as the ceramic-coated infrared burner, the radiant cup burner, and the radiant tube burner.

- The hybrid burner technology which takes advantage of the combined heat transfer effectiveness is the luminous flame burner. This burner is based on oil-gas cofiring technique and is designed to overcome the inherent disadvantage of gas-only firing which has lower luminosity compared to oil furnaces.

These burner developments resulted in a wide variety of burner equipment for many industrial applications. The application's performance depends on the selection and matching between designed burner performance and application requirements. It is noteworthy that state-of-the-art burners are sold as an integral part of boiler or furnace equipment and, as such, individual price per burner is not usually considered relevant to the marketing of the new gas equipment.

In the following section, the burner development experience for boilers and furnaces is presented.

5.1 Boiler Technology Needs

Boilers for heating of liquid and steam have user requirements which specify a design with:

1. Low NO_x emission to comply with the boiler emission standards
2. Stable combustion at excess air conditions
3. Wide turndown range for accommodating variation in the load
4. High heat transfer to liquid and steam

Two-Stage Burner

The first example of a low NO_x burner for boiler and dryer applications is the two-stage burner (the TS burner) which is depicted in Figure 7. The first stage of combustion takes place within the primary combustion chamber with fuel-rich conditions. The flame temperature in the primary combustion chamber is as high as 2500°F. The hot flame with the mixture of residual fuel and primary combustion product enters the second stage combustion zone which is under excess air condition. Reaction is completed within the secondary combustion zone and low NO_x product gas is generated. The fuel rich primary combustion chamber assures flame stability at excess air conditions in the secondary combustion zone.

The typical TS burner has the capacity in the range of 0.75 - 2.0 million Btu/hr. Stable turndown can be achieved up to 7:1. The burner diameter for an 0.8 million Btu/hr unit is approximately 14 inches, and the diameter of a 1.8 million Btu/hr unit is about 17 inches. Stable excess air range is reported to be between 5% to 20%.

Duct Burner

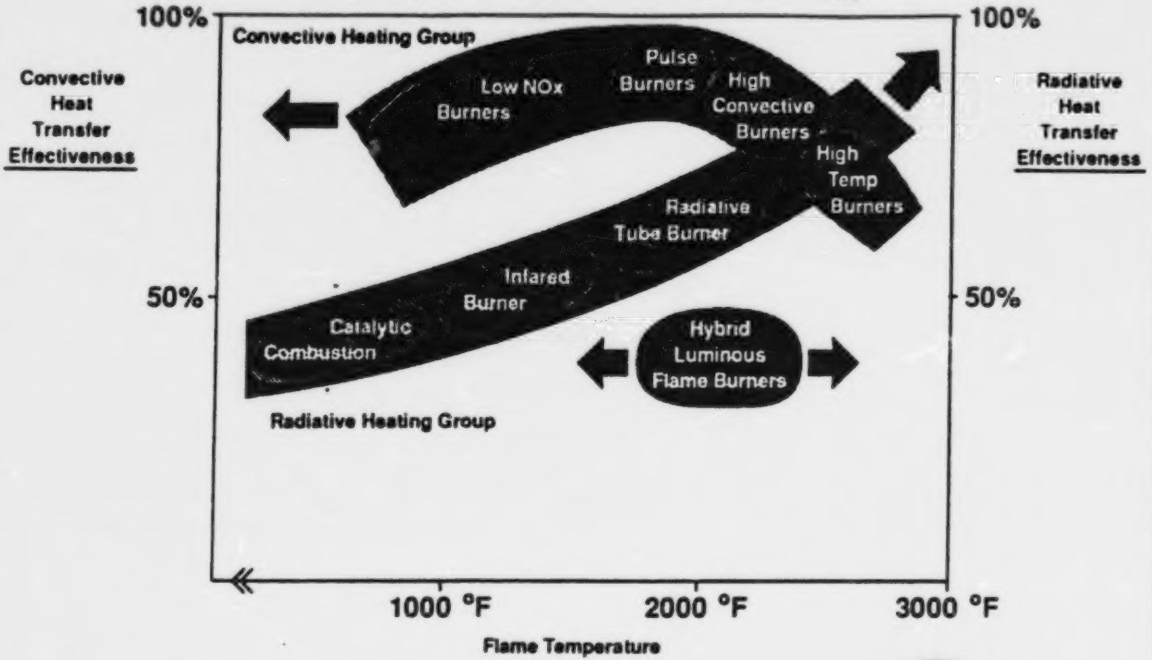
The duct burner for hot blast tempers and hot blast dryers is another type of excess air combustion equipment with a forced air system as illustrated in Figure 8. Typical output of the duct burner is in the range of 100,000 - 900,000 Btu/hr. The design blast temperature range is between 90° to 540°F. The excess air range of this burner is over 2.2 with a wide turndown ratio. Because of the wide blast temperature range, the system is designed to operate within the hot air duct for the blast temperatures and dryers. The burner width is 7 inches for the 170,000 Btu/hr unit and 14.5 inches for the 880,000 Btu/hr unit. A built-in forced air fan is capable of delivering the duct blast velocity of up to 100 ft/sec.

Submersible Burner

The submersible burner for liquid heating has been designed for relatively low temperature gradients of liquid heating, which is illustrated in Figure 9. Typically, in submersible burner applications or direct contact liquid heaters, there is a marked degradation in heating efficiency as the liquid temperature reaches the boiling point. The Japanese submersible burner has been designed for applications such as swimming pool heating and batch preheating of textile dye color. Advance features incorporated in the submersible burner are a large turndown ratio (of 10:1) and the baffle plate structure which prevents surface agitation of liquid and provides quiet operation. Aided by the wide turndown ratio, the burner can be operated to provide accurate temperature control which is important for textile dyeing and swimming pool maintenance.

Typical gas input capacities of the submersible burner range from 1 - 4 million Btu/hr. The diameter of the burner assembly range from 36 inches for the 1 million Btu/hr unit to 55 inches for the 4 million Btu/hr unit.

Summary of Industrial Burner Technology



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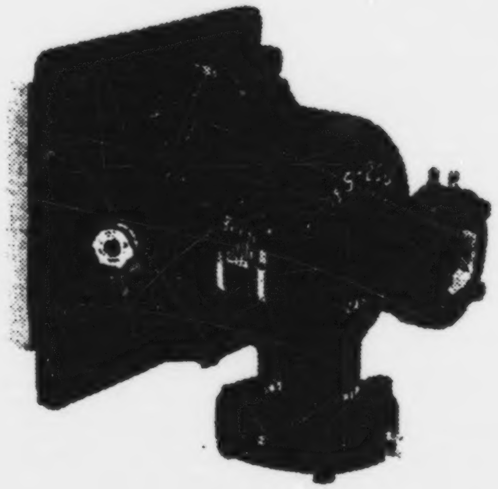
Two-Stage Burner
For Boilers and Dryers



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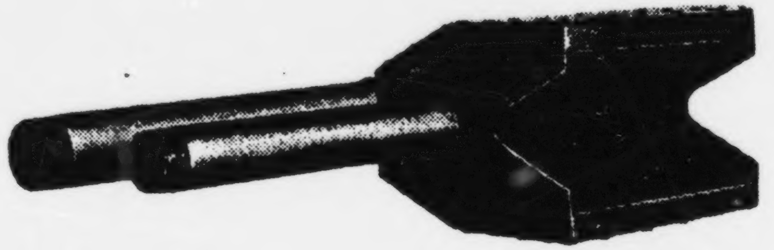
Duct Burner
For Hot Blast Dryers



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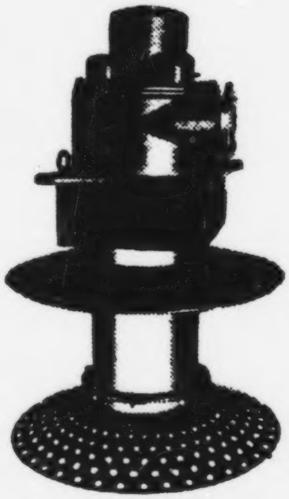
Submersible Burners
For Liquid Boilers



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5.3 Furnace Technology Needs

User specifications for furnace technology development are:

1. Flame width and length control
2. Low NOx emission to comply with furnace standards
3. Ease of process interface to provide quality improvement
4. Combination of convective and radiative heating enhancement
5. High convection burner applications

Flat Flame Burner

The flat flame burner is a typical example of the application of the Flexible Flame System (FFS) by which a wide variety of flame configurations can be obtained by simple modification to the burner design. The flat flame burner, as shown in Figure 10, has been designed to provide close-proximity heating to make the equipment more compact. Because the flame surface is more diffused and distributed over a wide area, a turndown ratio as high as 1:20 is achieved in a stable fashion. Since the burner is insensitive to the flame orientation, it can be mounted in any orientation. The wide turndown ratio also corresponds to a wide temperature range available.

There is a provision for preheating the combustion air to further improve the thermal efficiency of the equipment. The flat flame burner is used principally in areas of steel hardening and tempering furnaces, forging furnaces, annealing furnaces for copper and aluminum, and melting furnaces for metals with low melting temperatures.

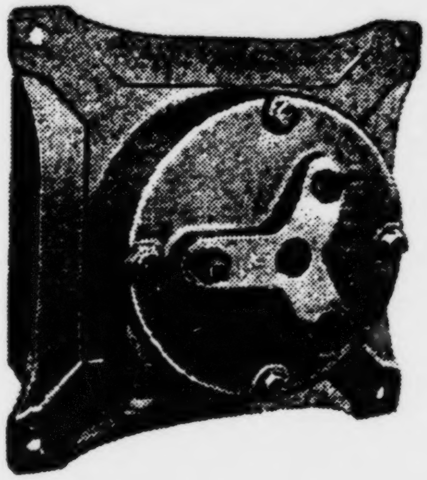
Typical input capacity of the flat flame burner ranges between 90,000 - 550,000 Btu/hr. The corresponding dimensions are 10-inch dia. and 17-inch diameter, respectively. The relatively low input capacity of these burners makes it more practical to use a number of burners in series at certain intervals in a given furnace. This is called the multi-port construction of furnaces, and provides excellent part load performance by appropriately selecting the number of burners to be in operation for a given load.

High Convection Burner

The high convection burner injects high temperature, high velocity gas into large annealing furnaces and melting furnaces. Basic burner geometry is shown in Figure 11. The design allows for combustion air preheating for improved thermal efficiency. The burner uses the end-mixing of air and fuel, and thus back firing can be effectively prevented.

Typical sizes range from 1 - 4 million Btu/hr. Experimental data indicate that for the 1-million Btu/hr unit, the injection jet temperatures ranges from 2900°F at the jet injection port to 540°F at 5 diameters away

Flat Flame Burner For Metal Treatment



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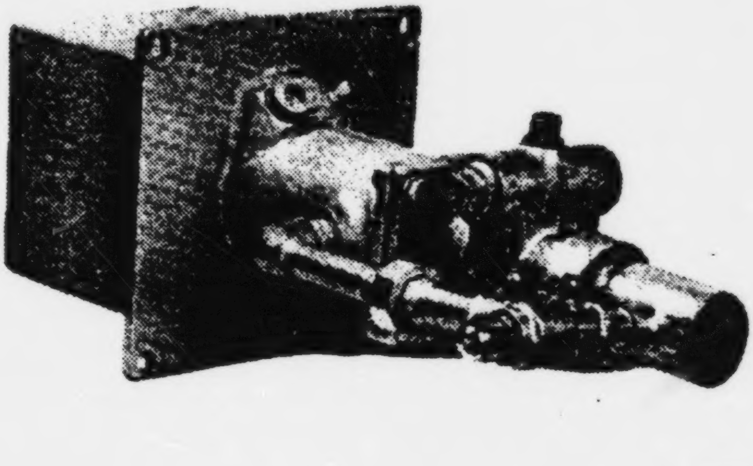
High Convection Burner For Annealing Furnaces



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from the port. A melting furnace employing the high convection burner places the nonferrous material on the bottom of the furnace just below the high heat flux jet.

Radiant Tube Burner

The radiant tube burner takes advantage of high radiative heat transfer from large surface area. Unlike the ceramic recuperator design from the U.S., the equipment shown in Figure 12 employs metal tube construction. Benefits of this design has been reported to be low cost construction, low combustion noise level, high radiative heat transfer, uniform temperature distribution over the surface, and ease of process interface. Principal applications can be found in gas cementation, non-oxidizing furnaces, and hot blast tempering.

Of the systems available today, input capacities range between 130,000 - 220,000 Btu/hr. The maximum surface temperature is limited at 2000°F. Effective radiative heat transfer area of the equipment range between 5.4 ft^2 for the 130,000 Btu/hr system and 7.8 ft^2 for the 220,000 Btu/hr. Stable air-to-fuel ratios range from 1.25 to 3.

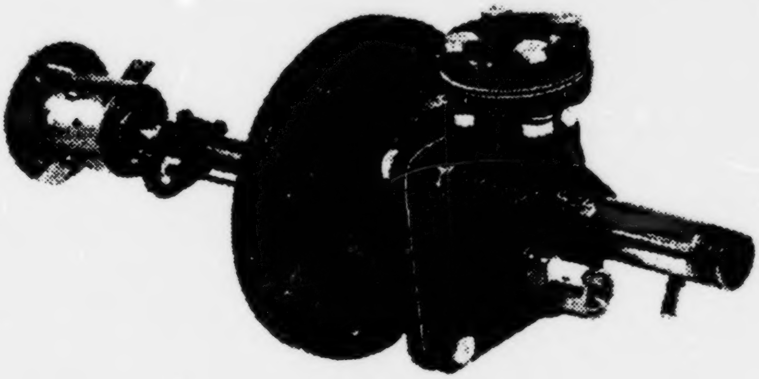
Oil-Gas Luminous Flame Burner

Typically in replacement of an oil furnace with gas burners, heating uniformity becomes a serious problem. This is due to the high luminosity of oil flame which contributes effectively to radiative heating of the furnace work area. However, a straight replacement of such oil burners with gas will cause a significant degradation of radiative heating because of low luminosity of gas flame. Low luminosity of the natural gas flame is, in turn, caused by the characteristic ratio of carbon to hydrogen atoms in methane (1:4) which is significantly lower than more complex hydrocarbon fuels.

In order to expedite replacement of oil by gas as a preferred fuel source in industrial furnace application, an oil-gas luminous flame burner was developed. The burner, illustrated in Figure 13, is based on two-stage combustion. Carbon particle (soot) is generated in the first stage with a low air ratio (approximately 10% of stoichiometry). Combustion is completed in the second stage by using secondary air injection. Luminosity of the flame is significantly increased due to the soot which has been seeded in the gas flame. Since the seed soot is consumed in the high luminous flame, particulate emission from the burner is effectively eliminated.

High radiative heat transfer effectiveness of the luminous flame burner is usually employed to preheat the combustion air in a typical design. With combustion air preheating of up to 800°F, improved energy efficiency as well as low NO_x production have been established. Current input design capacities range between 4 - 16 million Btu/hr. Aided by the supply pressure of gas, even heavy residual oil can be atomized by the fuel atomizer. The ratio of gas to oil can range between 20% to 100% gas (no oil) based on total burner output. Experimentally, it has been reported that the maximum radiative heat transfer occurs at a gas mixing ratio of 40% of total output.

Radiant Tube Burner For Nonoxidizing Furnaces



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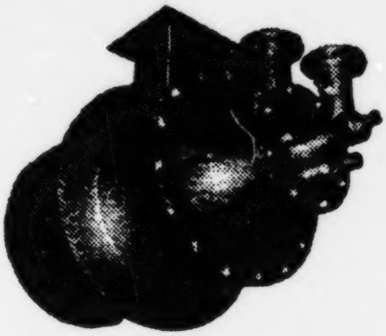
Oil - Gas Luminous Flame Burner

- Low NOx, SOx, Soot
- High Radiative Heating Effectiveness
- More Uniform Furnace Heating Over Length
- Use Radiative Preheating of Combustion Air



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The high luminous burner takes advantage of the hybrid heat transfer between convective and radiative mechanisms. Furthermore, the cofiring technique promotes reduction in NO_x and SO_x emissions from the oil-only burners. In applications for which simple replacement of oil furnace with gas burners is not desirable, the high luminous burner can provide an economic alternative.

6. FUTURE TRENDS

Overall, the emphases of the recent advances in Japanese industrial gas technology are on low NO_x burners, Flexible Flame Systems, high luminosity burners, and radiative tube burners. Technology drivers appear to be the economic and regulatory situations which require industrial energy conversion equipment to be efficient and pollution free and to increase plant productivity. The approach the Japanese gas industry has taken is to develop a number of burners with different flame configurations, flame temperatures, convective and radiative heating combinations, etc. Furthermore, the relatively small capacity of individual burner output is apparent.

This approach has been extremely successful in Japan because multiple burner boilers and furnaces allow for better turndown capability and ease of process interface over a wide range of factory loads. By having a wide variety of burner options, users are able to select from a combination of burner size, flame characteristics, and radiative and convective heating groups. For most users, the availability of many different types of burners and associated equipment is unknown. The gas industry marketing effort has focused on individually approaching key industrial accounts to introduce the depth of available technology and to develop alternative options for economic analysis by the user.

The Japanese experience has been successful in developing "process-specific" and "site-specific" equipment design through selection from a large line of gas burners, engines, and chillers. The outlook of future applicability of offshore technology in the North American market is quite positive in that today's industrial customers are geared to compete in the flexible manufacturing process which can handle a large swing in factory output capacity. Thus, the successful experience gained through offshore technology development will provide a great opportunity to design industrial gas equipment based on responsiveness to user needs. As many of the leading manufacturers from Japan will be establishing the North American factory base, so too will Japanese gas equipment become available. This possibility of the future influence of Japanese technology in the North American gas equipment industry should serve as an impetus to upgrade domestic technology base in order to respond to the needs of key domestic gas customers.

SECTION

IV

DEVELOPMENT AND APPLICATION
OF THE
ICCS COMBUSTION CONTROLLER

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B.C.
13 - 14 April, 1988

by Robert E. Sovik and Douglas E. Deery
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Bob has been concerned with various aspects of instrumentation for over 15 years and joined Bailey Controls in 1984. He is now involved in the International Operations of this company.

Bailey Controls in Canada has been active in Boiler Controls since 1921 and is a major supplier of instruments and control systems for boilers and processing plants.

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DEVELOPMENT AND APPLICATION
OF THE
ICCS COMBUSTION CONTROLLER

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Presented at the
Industrial Gas Technology Conference
Vancouver, British Columbia Canada
April 13, 1988

DEVELOPMENT AND APPLICATION
OF THE
ICCS COMBUSTION CONTROLLER

ABSTRACT

The need for a low cost combustion controller for low-end gas users has existed for sometime, but until recently the necessary technologies were not available to allow the cost effective development of such a controller. New developments in materials, manufacturing, and combustion control technologies are available and are causing a radical change in the way combustion controls for small, gas-fired industrial furnaces, boilers, and heaters are being implemented.

These significant advances in technologies include sensor, electronic, and packaging improvements.

Sensors that can measure the constituents in the flue gas stream of a combustion process (e.g., O2 and CO) can be made less expensive, more reliable, and require less effort in calibration and maintenance.

The use of digital technology in the electronics area has substantially reduced electronics costs while increasing functional capability and reliability.

Lower fabrication costs and packaging techniques are now available because of the use of new materials and manufacturing processes. Combining these technologies can result in a combustion controller that is significantly lower in cost than existing devices and therefore applicable to a much wider range of applications.

INTRODUCTION

Realizing the use of natural gas in the process industries was coming under increasing pressure from competing energy sources, the Gas Research Institute (better known to some as G.R.I.) and the Bailey Controls Company initiated a program to develop a low cost combustion controller for the natural gas user. The end product of this development program is a low cost Industrial Combustion Control System (ICCS) which provides significant benefits to the natural gas industry in general and the end user (rate-payer) in particular.

DEVELOPMENT OBJECTIVES

The key objectives of this joint program were to develop a low cost combustion controller for industrial natural gas fired processes which would provide:

- Improvements in efficiency of gas combustion processes
- Improvements in the quality of the end user product
- Increased safety of the combustion process operation

Many studies have shown that the efficiency of a combustion process depends on operating it at the proper fuel/air ratio. The closer the ratio is to stoichiometric combustion conditions, the higher the resulting efficiency. By measuring the values of oxygen (O2) and carbon monoxide (CO) in the exhaust gases, the low cost combustion controller ensures that the process is operated at or near the optimum efficiency.

The relationship between fuel/air ratio and efficiency is illustrated in Figure 1, which shows the effect of using excess air (beyond stoichiometric) on the measured values of O2 and CO and the resulting process efficiency.

Too much excess air decreases efficiency because the energy expended to heat this air is wasted up the stack. Too little excess air results in fuel being unburned and also wasted up the stack. Some level of excess air is required to completely burn the fuel because of imperfect fuel/air mixing conditions in any commercial burner.

Note from Figure 1 that the value of measured O2 increases as the excess air value is increased above the optimum. Also, the value of measured CO increases as the excess air value is decreased below the optimum.

Depending on the specific combustion process, the optimum value of O2 is in the range of 1 to 3 % and the optimum value of CO is in the range of 200 to 400 parts per million (ppm). Therefore, a combustion controller that provides measurements of O2 and CO in these ranges can ensure that the proper fuel/air ratio is maintained in the face of disturbances such as changes in fuel heating value or deterioration of burners.

PROCESS IMPROVEMENTS

In the low cost combustion controller application, both O2 and CO measurements are required. The reason for this is that some processes must be run slightly on the oxidizing side of stoichiometric to protect the manufactured product from the by-products of fuel-rich combustion. Other processes must be run slightly on the reducing side to keep the product from oxidizing. Having both measurements allows the controller to run the process at maximum efficiency while maintaining the atmosphere that is appropriate for the manufactured product.

The amount of improvement in efficiency (and reduction in fuel use) that can be obtained for a particular process depends on two factors:

- the current fuel/air ratio at which the process is being operated.
- the temperature of the exhaust gases going up the stack.

In most processes, efficiency is lost because the fuel/air ratio is too lean (too much excess air). The effect of this type of loss is shown in Figure 2 in which the percent fuel required is plotted as a function of excess oxygen for several values of exhaust gas temperature.

It is clear from this figure that the potential savings are greatest in the case of high-temperature processes. For example, a reduction of excess oxygen from 6% to 5% for a process with 1500 oF exhaust gases produces a fuel savings of over 5%. For a heating process consuming 50 MCF per year, this translates into a dollar savings of $12,500 per year. Even if the fuel savings for this size process were only 2%, this would still result in a savings of $5,000 per year. The potential savings for processes that are run on the reducing side of stoichiometric are even more dramatic, since the efficiency curve drops off faster on that side rather than of the oxidizing side for an equal change in fuel/air ratio.

QUALITY IMPROVEMENTS

By providing more accurate control of the fuel/air ratio in the combustion process, the controller will improve the quality and reduce the cost of the end product manufactured using the process.

Until recently, it is not cost-effective to measure the O2 and Co concentrations in the vicinity of the product; as a result, the operators of the process select a fuel/air ratio based on trial-and-error (resulting in much scrap product) supplemented occasionally by sampling the combustion gases using portable analyzers.

Of course, this portable sampling approach does not automatically compensate for hour-to-hour variations in operating conditions (e.g., changes in product throughput rates, variation in BTU of fuels) that might cause variation in product quality beyond the allowable specification range. Some examples of this deterioration of product quality due to changing combustion conditions are as follows:

1. In the iron and steel industry, slab reheating furnaces are supposed to be run at about 1% excess oxygen to avoid scaling on the slabs. Deviations from this level now cause significant product waste and rework.

2. In aluminum die cast operations, the melting furnace is supposed to be operated slightly on the oxidizing side. If the environment becomes too fuel-rich the aluminum can absorb hydrogen, which causes bubbles in the resulting castings. On the other hand, if the environment is too fuel-lean, the resulting oxidation of the aluminum causes waste (up to 3 - 4%) and requires the use of additional fluxes to clean the castings.

3. Heat treating furnaces in the iron and steel and copper/bronze industries must also be run at or near stoichiometric conditions to avoid oxidation and resultant waste of the metals.

Thus as can be demonstrated, a low cost combustion controller can allow significant product savings to be achieved, even in small processes.

SAFETY IMPROVEMENTS

The final benefit of the low cost combustion controller is that its' improved measurements, diagnostics, and fail-safe control capabilities will ensure that the processes equipped with the controller are run more safely. Some examples of improved safety of operation are as follows:

1. Existing combustion controllers have failure modes that are potentially unsafe; if, for example, the air flow controller fails and shuts the air damper. The new controller has a fail-safe design feature that allows the user to specify the default value of the control outputs if there is a controller failure. In this case, the user would specify

that the fuel valve be closed and the air damper opened in case of a failure.

2. The self-diagnostic features of the new controller detect internal problems quickly, alert the operator, and put the combustion process in a safe state. For example, the "intelligence" of the microprocessor-based controller allows detection of a failed O2 or CO sensor. This capability improves the safety of process operation during failure conditions.

3. The logic capabilities of the low cost combustion controller permit the implementation of process startup procedures that are safer than before. For example, the controller can be interfaced and configured so that it goes through a timed air purge process before it allows the burner lightoff sequence to begin, thus ensuring that no combustible gases are present in the furnace or boiler at lightoff.

Via these approaches and others, the safety features now available using expensive control equipment are made affordable through the use of low cost combustion controller.

SENSOR DEVELOPMENT

To achieve the benefits discussed, the development program focused on the three key areas of sensors, electronics and packaging. All three of these are important to the success of developing a low cost combustion controller but the sensor area is probably the most critical. The sensors are required to measure the key process variables that relate to combustion efficiency. The strategy of using a combination oxygen and carbon monoxide analyzer to provide the vital information needed for optimum combustion control was employed (Figure 3).

The technology most widely favored for measuring O2 content is the use of the solidstate electrochemical zirconium oxide cell. The sensor cell is comprised of a membrane having specific ion conductivity for oxygen at temperatures above 1100°F. Zirconium dioxide, with a stabilizer, is a material used for such cells.

This sensor cell has a characteristic that gives it a significant safety-related advantage over other methods: it indicates the "net" oxygen that remains in the sample after any combustibles (such as methane) are burned. Therefore, if any combustibles are present, it will cause the control system to increase the air flow to compensate. Another advantage is as the O2 concentration decreases, the sensor output increases to produce a more accurate and repeatable signal.

These cells require that the sensor temperature be reasonably high before the electrolyte starts to conduct an adequate supply of oxygen ions (available from the reference air supply). The output of the sensor is directly related to the product of the absolute temperature and to the logarithmic difference in the partial pressures of the O2 concentration across the membrane. Specific work was directed toward the new O2 sensors which have been introduced in the automotive industry. These sensors have been shown to be quite adequate when used under the right conditions of voltage, current, and temperature.

There are several different methods for determining the CO content in a sample stream; however, there is only one which easily lends itself to be incorporated into a single sensor assembly with an O2 sensor. The technique is based upon catalytic technology and has been used by the Bailey Controls Company for the last eight years. The exact measuring principle uses a proprietary catalyst which makes it more specific to carbon monoxide but does allow it to respond to other combustibles (methane, ethane, propane, etc.) if they are present.

ELECTRONICS DEVELOPMENT

The electronics package for the low cost controller consists of the microcomputer and support electronics, user interface hardware, input/output circuitry, sensor interface, and power circuitry (Figure 4).

A microprocessor based architecture was chosen to provide the extensive computations and overall intelligence required to perform combustion control. The microprocessor is responsible for continuously monitoring the controller's keyboard and updating its Liquid Crystal Display (LCD). The LCD display chosen, a 20 character by 4 line model, is used to display relevant process conditions and operating instructions.

A portion of the electronics section handles the inputs form the sensors. This information is "processed" by solving proven control configurations or equations and comparing the results with a known standard or set-point. The differences are translated into output signals to correct the deficiency. The microprocessor, controlled by the system's firmware, performs the complex calculations required to efficiently control a combustion process.

The microprocessor chosen for the controller is the Motorola 68HC11, a recently developed, highly integrated device. Included on the chip are significant amounts of Read Only Memory (ROM), Random Access Memory (RAM), and Electrically Erasable Programmable Read Only Memory (EEPROM), and a fairly sophisticated timer system.

Due to the high degree of integration of the 68HC11, the complexity of the low cost combustion controller's electronics is reduced. Many of the functions which would have been performed by a large number of Integrated Circuits (IC's) or a semi-custom chip are now performed by one standard part.

As electronics technology has developed, more and more functions have been incorporated into smaller spaces and fewer circuits. This has led to fewer parts and lower cost to achieve more operations. In direct relation to the lower number of parts involved electronically, is the increase in product reliability which impacts overall cost in the longrun.

Another definite advantage of higher integrated circuitry such as micro-processors and micro-controllers is the ability for them to calibrate themselves. The next logical extension would be auto-calibration and set-up for the entire system. This can be done as often as needed, with no external interface required.

PACKAGING DEVELOPMENT

In the area of combustion control systems, packaging plays a very important role. Here, packaging consists of much more than just the two external enclosures (Figure 5). The housing for the sensors, along with the manifolding passages are a small part of packaging. A better description of the packaging function could be system integration which combines the various parts into a working integrated system.

The electronics assembly has a membrane keyboard and a 20 character 4 line liquid crystal display on the cover. This comprises the user interface and allows communication with the outside world, consisting of current system values and operator entered set points. This information is visible on the screen and can be scrolled for viewing.

Under the sensor assembly cover, we have a complex system of manifolds, flanges and labyrinths. The passages direct the incoming sample gases to their proper sensing areas, while the labyrinth is used to stabilize the temperature of this sample before passing over the sensors.

This sample is drawn through the system by an aspirator that consumes low volumes of instrument or shop air. Heaters are located a various points to assure that the metal parts are above the dew point of the sample gas, and thus prevent condensation and introduction of moisture to the system.

After the sample is analyzed, it passes through the aspirator and is dumped back into the flue, insuring safe handling of the sample.

The two enclosures communicate through a prefabricated cable. Signals include the millivolt outputs from the oxygen and carbon monoxide sensors as well as power to the various cartridge heaters and thermocouple to establish control.

The major direction of packaging development was the incorporation of low cost materials which meet the engineering needs as well as lending themselves to automated manufacturing and assembly procedures.

To accomplish these tasks in the most cost effective manner, various materials were investigated. Some of the materials used are cintered powdered metals, ceramics, and high temperature plastics for the sensor manifold assembly.

Advances in manufacturing process technologies in the areas of casting and molding have allowed the integration of many functions into as few parts as possible. This feature, along with new materials and economics of scale, facilitate production of quality, low cost parts.

CONCLUSION

The need for a low cost combustion controller for the low end natural gas users has existed for sometime, but until recently the technologies were not present to allow the cost effective development of such a device. For this device to have a significant impact on the usage and cost of natural gas it must have a one yaar payback period over a wide range of gas uses and equipment sizes. With this in mind, Bailey Controls initiated a development program for a low cost combustion controller. This program addresses the need of the marketplace from a total system integration perspective. Using the latest state-of-the-art technologies in sensors, electronics, and packaging, the feasibility of such a device has been shown.

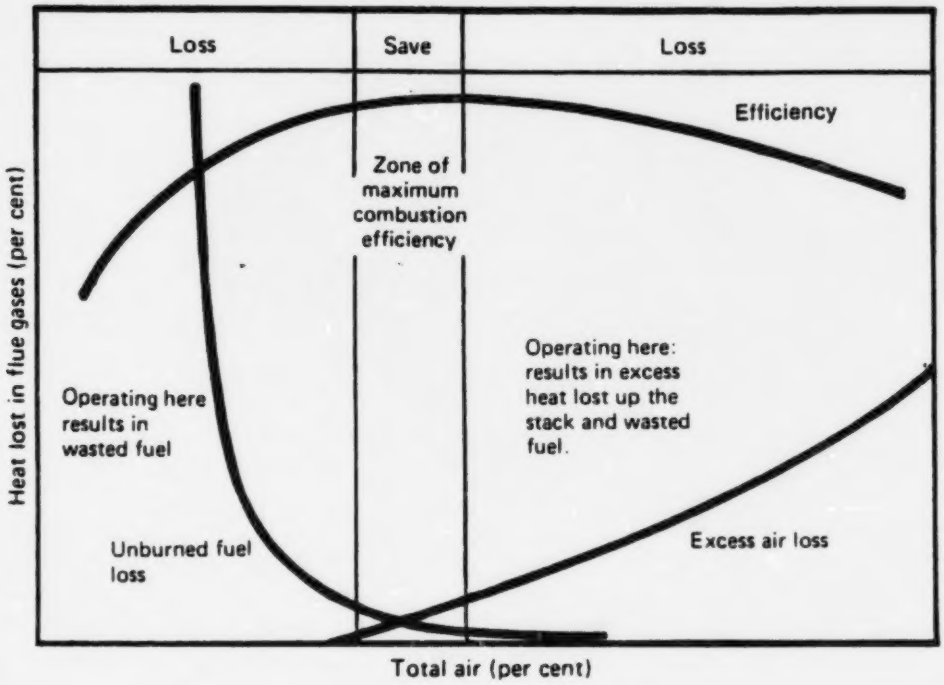

FIGURE 1 — Maximizing Combustion Efficiency

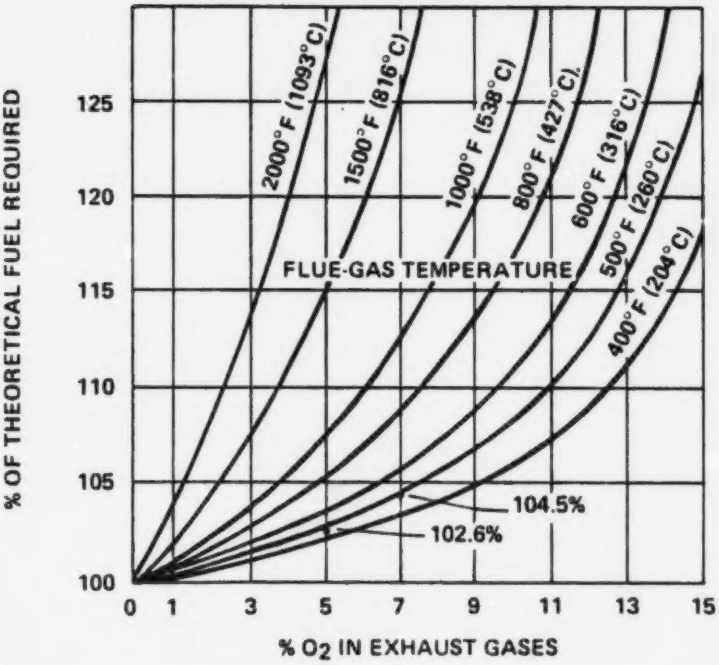

FIGURE 2 — Efficiency Losses Due to Excess Air

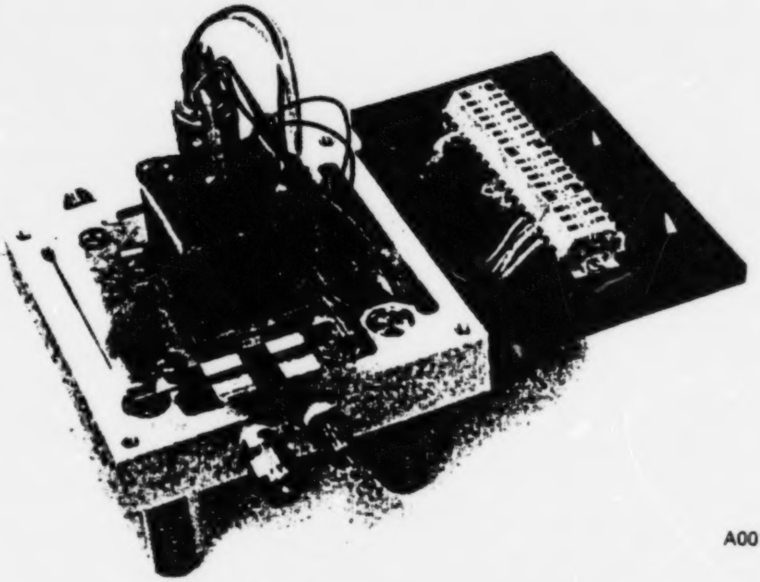

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FIGURE 3 — Oxygen and CO Sensor Test Assembly

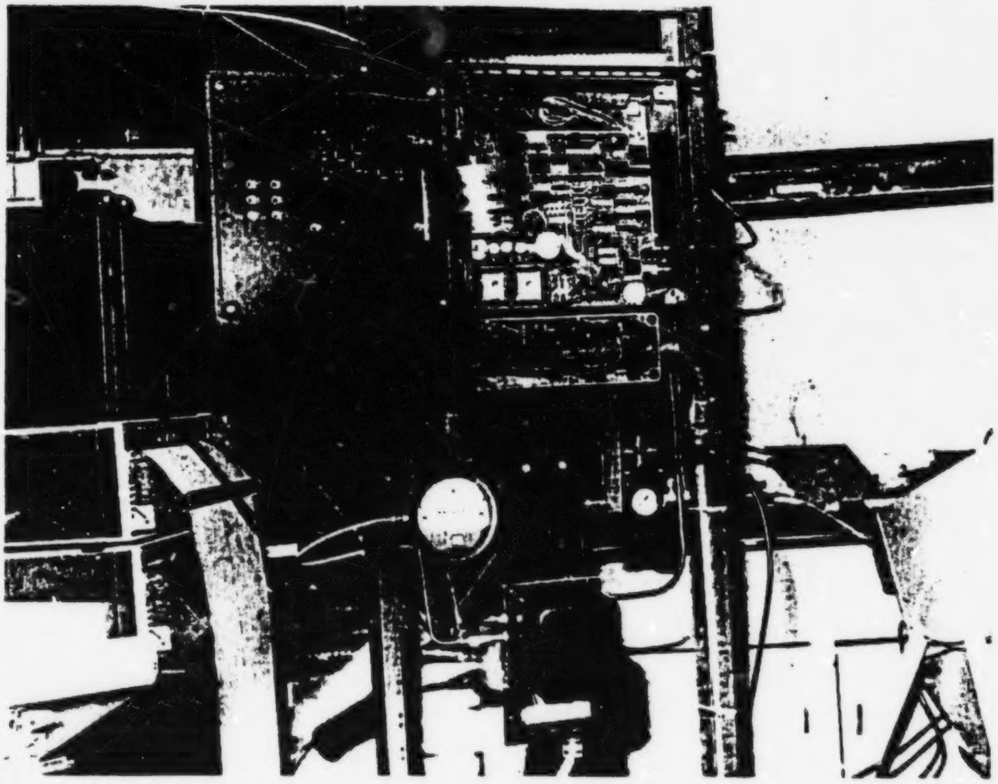

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FIGURE 4 — Electronics Assembly

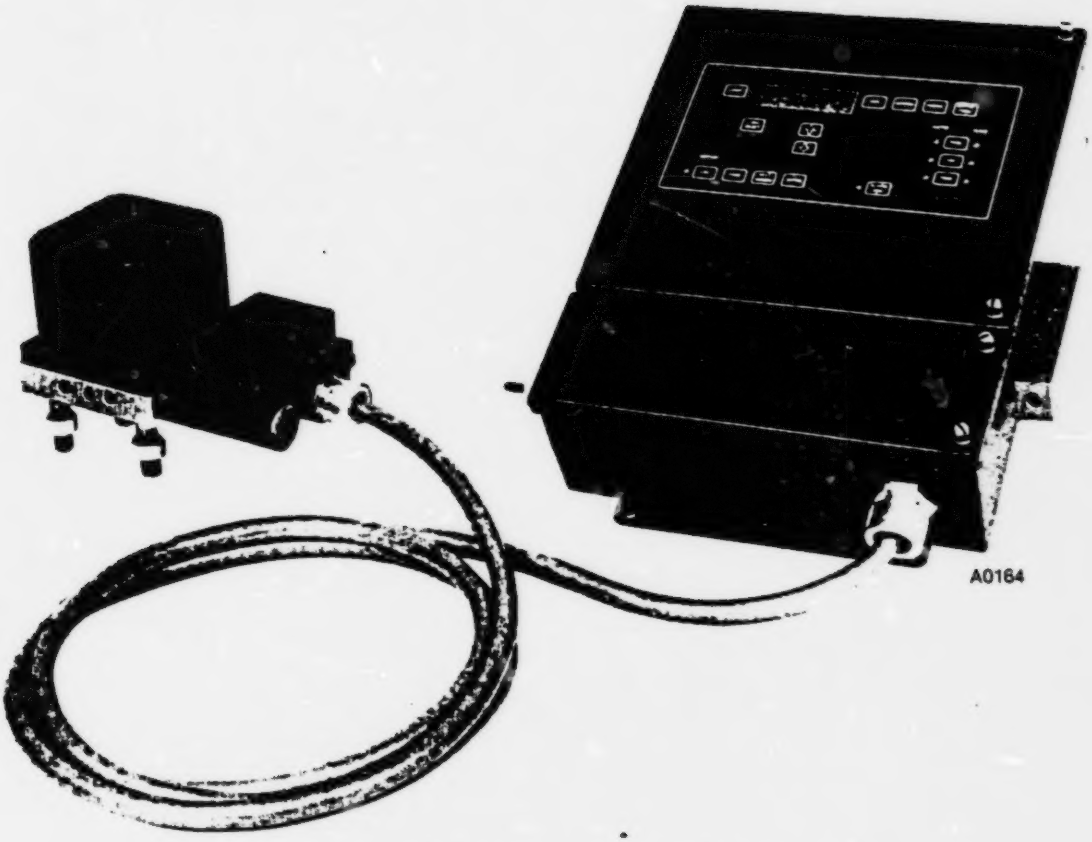

FIGURE 5 — Industrial Combustion Control System (ICCS)

SECTION

V

FLUE GAS HEAT RECOVERY

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B.C.
13 - 14 April, 1988

by Paul Willis
B.C. Hydro

ABOUT THE SPEAKER

Paul Willis has been with B.C. Hydro for 14 years, working as a mechanical design engineer, project engineer and energy management engineer.

As an energy management engineer for the last 4 years he has been involved in advising industry on energy use. His speciality is industrial process heating. His role as an advisor involves plant visits, writing technical information documents, and taking part ih and orgainizing technical seminars such as this conference.

He graduated from the University of Waterloo in 1970 with a Bachelor of Science degree in Mechanical Engineering. His initial work experience was with Babcock & Wilcox as a boiler design engineer.

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FLUE GAS HEAT RECOVERY

Introduction

The objective of this paper is to examine the energy balance factors which govern the flue gas heat recovery alternatives available in a boiler system. Heat recovered from a waste heat stream has to be returned to a system input stream that is at a lower temperature. This basic principle is explained with respect to a typical 20,000 lb/h steam boiler system.

Boiler Heat Recovery Potential

Table 1 indicates the efficiency of a gas fired boiler operating with no heat recovery and a similar boiler operating with the maximum amount of heat recovery.

Boiler With No Heat Recovery		Boiler With Maximum Heat Recovery	
Excess Air -	10 %	Excess Air -	10 %
Stack Temp. -	520 F	Stack Temp. -	80 F
Dry Gas Loss -	8.2 %	Dry Gas Loss -	.2 %
Moist. Loss -	11.9 %	Moist. Loss -	1.5 %
Radiation Loss -	1.0 %	Radiation Loss -	1.0 %
Blowdown Loss -	.6 %	Blowdown Loss -	.6 %
Efficiency -	78.3 %	Efficiency -	96.6 %

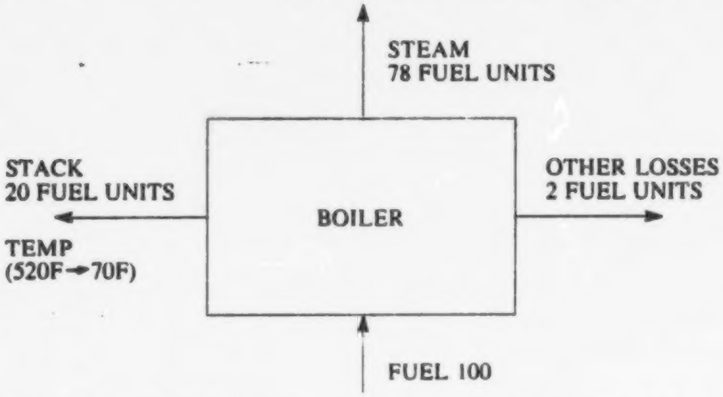
BOILER EFFICIENCY CALCULATIONS

TABLE I

From this table it can be seen that the difference in efficiency between a boiler operating with the maximum amount of heat recovery to a boiler operating with no heat recovery is approximately 18 percentage points. For a 20,000 lb/h boiler burning 4 $ per million Btu natural gas and operating 5,000 hours per year, these 18 percentage points represent an annual savings of $ 100,000.

The maximum heat recovery example is based on a stack temperature of 80 F and assumes an ambient temperature of 70 F. In the no heat recovery example, it is assumed that the boiler is producing 250 PSIA steam in which case the stack temperature would be approximately 520 F.

Figure 1 illustrates in very simplified terms the total heat potential available. For every 100 units of fuel burned 20 fuel units are wasted in the stack flue gases if no heat recovery is used.



SIMPLIFIED HEAT BALANCE FOR BOILER WITH NO HEAT RECOVERY

FIGURE 1

Heat Recovery With Economizer

With most boiler systems where a deaerator is used the boiler feedwater temperature is usually in the order of 220 F. The saturation temperature for water at 250 PSIA is 400 F, therefore theoretically the boiler feedwater could be heated from 220 F to 400 F prior to entering the boiling surfaces. Due to considerations for upset conditions, the practical maximum feedwater temperature is in the order of 350 F. For our typical 20,000 lb/h unit operating at 250 PSIA, the energy required to heat this feedwater is approximately 10 fuel units for every 100 fuel units burned. Figure 2 compares this heating requirement with the waste heat going up the stack.

It is important to note when matching a waste heat source with a heating requirement consideration has to be given to the amount of heat in both cases and to the temperature range in both cases. From figure 2, it can be seen that an economizer used to heat the feedwater could only reduce the stack temperature to around 250 F because the coldest feedwater is 220 F. The heat saved by cooling the flue gas from 520 to 250 F is 6 fuel units. These 6 units will heat the feedwater from 220 to 300 F as shown on figure 2 and for boilers in this pressure range usually economizers will be operating at these conditions; cooling the flue gas from 520 to 250 F and heating the feedwater from 220 to 300 F.

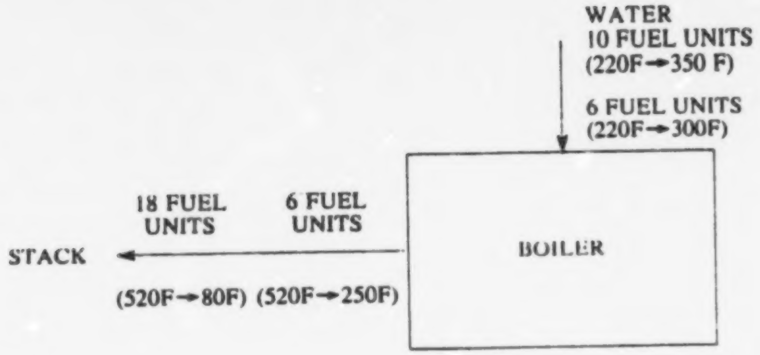


FIGURE 2

HEAT BALANCE FOR BOILER WITH ECONOMIZER

Corrosion conditions are also important. It so happens that an economizer with a minimum water temperature of 220 F and gas temperature of 250 F is out of the severe corrosion region because even during upset conditions there shouldn't be any condensate on the economizer surfaces.

Heat Recovery With Air Heater

Figure 3 illustrates the situation if an air heater is used. With respect to the air heating requirement, it would be practical to heat the air from 70 to 500 F. This temperature range also matches the desired temperature cooling range of the flue gases. However, it should be noted that the quantities of energy don't match. There is 20 units of waste heat and only 9 units of air heating required.

Corrosion conditions are usually important in selecting the design conditions of air heaters. Conventional air heaters are not designed to cool the flue gas below 250 F, because lower temperatures will lead to condensate on the metal surfaces particularly during upset conditions. Therefore as is shown in figure 3, a typical air heater would cool the stack gases from 520 to 250 F and would heat the incoming air from 70 to 340 F. It is possible to use more expensive corrosion resistant surfaces in order to reduce the stack temperatures below 250 F, however the economics usually does not favor this, for air heating alone, because their is only another 3 possible waste units that could be used by the air stream.

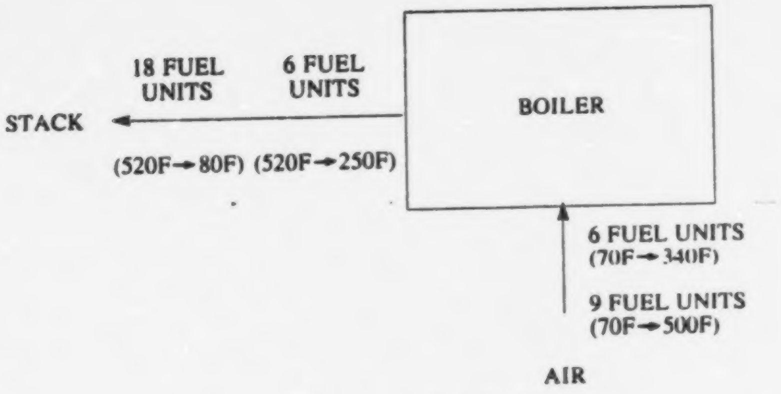

18 FUEL UNITS
6 FUEL UNITS
STACK
BOILER
(520F→80F)
(520F→250F)
6 FUEL UNITS
(70F→340F)
9 FUEL UNITS
(70F→500F)
AIR

FIGURE 3

HEAT BALANCE FOR BOILER WITH AIR HEATER

Low Temperature Heat Recovery

From the above it can be seen that for this size of boiler, an economizer or an air heater can recover approximately 6 of the potential 18 fuel units of waste heat available. The remaining 12 units exist in the flue gas between the temperatures of 250 and 80 F and, actually most of these 12 units, around 10, exist within the water vapour portion of the stack gases and can only be recovered by condensing this moisture at temperatures below 100 F.

A number of companies are now manufacturing condensing heat exchangers. However, to be able to make use of them it is necessary to find a heating requirement in the 80 to 120 F range. Some possibilities are as follows:

Item	Desired Heating Range
Boiler Makeup Water	50 F to 180 F
Domestic Hot Water	50 F to 130 F
Swimming Pools	50 F to 100 F
Irrigation Water	50 F to 80 F
Heavy Oil	Ambient to 180 F

Another possibility for recovering this low grade waste heat if none of these low temperature heating requirements are available is to use a heat pump to upgrade the temperature. A heat pump for example can quite efficiently recover waste heat from a 100 F source and deliver it to a 150 F requirement.

Conclusions

The most important rule in heat recovery is that to recover energy from a waste heat source a heating requirement must exist at a lower temperature. At first glance it might appear that a heat pump circumvents this rule but actually if a heat pump is used a low temperature heating requirement is produced in the evaporator section of the heat pump cycle.

Improved materials technology allows us to recover waste heat from low temperature sources but to utilize this recovered energy we must be innovative in finding low temperature heating requirements.

SECTION

VI

NEW GAS TECHNOLOGIES FOR LOW TEMPERATURE INDUSTRIAL HEATING APPLICATIONS

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Jacques Martrès
Gaz de France

ABOUT THE SPEAKER:

Jacques Martrès received his Engineering degree in 1978 and in 1982 a M.B.A. degree, specializing in marketing and management.

In 1983, Mr. Martrès was awarded the Certificate of the French Institute of Energy on the Optimum Utilization of Energy in Industry.

Jacques joined Gaz de France in 1978 where he was concerned initially in natural gas pipeline design and construction.

Since 1982 he has been head of the high temperature sector, responsible for gas sales to industry involving metallurgy, glass, ceramics and space heating, pilot plant operations, research market studies, etc.

In addition, Jacques is a professor at the College for the Application of Gas Technology in Paris.

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INTRODUCTION :

The heating of low-temperature industrial baths (below 100 °C) is responsible for a substantial amount of energy consumption in industry. Most often, this heat requirement is met by centralized production of steam and the use of exchangers which are located at the points of utilization.

However, the need to save energy has given rise to the development of decentralized solutions, which are very well suited to the use of gas fuels.

There are two processes which make possible "in situ" heating of industrial baths by natural gas :

- *direct heating by submerged combustion which is characterized by the absence of an exchange wall. There is maximum heating efficiency, but the application of this process is limited by the temperature and nature of the products to be heated ;*
- *indirect heating via compact immersion tubes or exchangers which are characterized by compactness and good heating efficiency.*

The different processes of industrial liquid heating analyzed here are characterized by :

- *__their location__ : they are set up as close as possible to the bath to be heated, and if possible, in the liquid (thus eliminating losses due to energy distribution) ;*
- *__their operation__ : these processes make it possible to quick meet demands for heat independently of the operation of other factory equipment ;*
- *__their performance__ : due to their location and operation, these processes enable good heating efficiency : 85 % to 100 % on NCV. (Net or Low Calorific Value.)*

I NATURAL GAS COMBUSTION :

I.1 Chemical reaction :

The combustion of natural gas is an oxidation reaction under which natural gas and air are combined into carbon dioxide, water vapor, nitrogen and excess air.

Heating processes using natural gas consist in extracting heat from the combustion products and transferring it to the fluid to be heated.

In order to obtain good thermal efficiency, the heat content of the combustion products vented to atmosphere must be as low as possible. To this end, the weight of flue gases is controlled through the amount of excess air and the flue gases are cooled following a maximum heat transfer.

I.2 Gross Calorific Value and Net Calorific Value :

By cooling down the flue gases below 60 °C, the heat of condensation contained in the water vapor generated during combustion is recovered. This amount of heat represents nearly 11 % of the total energy content of natural gas. The gross calorific value (g.c.v.) takes into account this amount of latent heat whereas the net calorific value (n.c.v.) does not.

Heating liquids by direct contact with the combustion products therefore has an excellent efficiency since the temperature of the flue gases can be reduced to a minimum value and the water vapor resulting from combustion can even condense where the bath temperature is not too high.

II DIRECT HEATING OF LIQUIDS : (ECS)

II.1 How Direct Heating Works :

Direct heating of liquids consists in bringing into contact the liquid to be heated and the combustion products :

- either by spraying the liquid into the combustion products
- or by bubbling the combustion products through the liquid itself (ECS).

In both cases, the heat exchange areas involved are very large since the total area of the bubbles from 1 cubic meter of combustion products is about 1.000 m^2.

In addition, the heat transfer is optimum due to direct contact and the absence of any exchanger wall. The transfer of heat to the liquid is further enhanced by the movement of both fluids. In the case of the combustion products bubbling through bath, the combustion products must travel up 50 cm only before the temperatures of the liquid and the combustion products are equal.

II.2 Impact on Liquid Properties :

Natural gas is a clean fuel and so are the combustion products since no sulfur compounds are contained. In the long run, carbon dioxide and nitrogen oxide traces slightly acidify the liquids which are directly heated.

In practice however, this acidification should be taken into account only when strongly alkaline baths are heated. In most cases acidification is negligible compared with the deterioration due to the use of the bath in a specific process.

II.3 Description : (Fig. 1)

The combustion products are sent into the liquid to be heated through a manifold of a suitable shape.

The combustion products leave the liquid surface at the same temperature as the bath. The combustion chamber is cooled by combustion air in order not to bring the liquid in contact with too hot a wall, which would have an impact on the flame property.

The control is either of the modulating or of the on-off type. In both cases, it is possible to obtain the required bath temperature with a high degree of accuracy.

This type of burner can be mounted into the tanks without any openings in the side walls. They can be easily removed so that it is not necessary to empty the tank.

This is a plus where maintenance is to be carried out. This type of burner can also be operated on a by-pass to the main tank.

One of the major advantages of this technique is that there is no thermal inertia since the temperature of the bath increases as soon as heat is introduced to the system. Since natural gas is a concentrated souce of energy, heating times are cut down to a minimum.

Alternatively, this type of burner may directly heat a thermal fluid from which heat is transferred to the working fluid through an exchanger.

II.4 Performance Data : (Fig. 2 & 3)

The combustion products leave the liquid at the same temperature as the bath :

- where the temperature is below 60 °C, part of the water vapor generated during combustion condenses and the efficiency is greater than 100 % (n.c.v.) ;
- between 60 and 70 °C, the efficiency hovers around 90 % (n.c.v.) ;
- above 70 °C, the efficiency drops since the water vapor does not condense any longer ; on the contrary, the bath water evaporates ; however, this can be used where the bath is to be concentrated.

The efficiency of direct liquid heating is at least 30 % higher than with a conventional heating system using steam where the temperature do not exceed 60 °C. This relationship holds up to 70°C.

II.5 Applications :

II.5.a Mechanical industries :

Surface treating (degreasing, phosphatizing, etc.) are ideally suited to direct heating :

- The bath temperatures are relatively low and most often below 60°C since bath suppliers more frequently offer efficient systems requiring low temperatures;
- The bath volumes may be large - favouring the high heat release and the flexibility of direct heating;

- The type of burner described above is characterized by ease of service compared to other technologies since the burners are easily removable and no exchangers are necessary.

.II.5.b Food Processing :

Dairies : washing plants can use either technique : medium temperature acid baths as well as water used for washina can be heated directly. Soda baths should however be indirectly heated through gas fired exchangers.

Slaughterhouses : tests and field experiments with a hog scalding vessel have shown that direct heating is compatible.

Breweries : the direct heating of liquids used in pasteurizers and bottle washing machines are under study. Wine can also be heated using this technique prior to filling.

II.5.c Textile Industry :

Direct heating should be suitable for baths like those used for bleaching. This technique is alread used for process water heating.

II.5.d Other Applications :

Direct heating can be used for many other applications where process hot water is needed eg. for mixing concrete, for washing, for heating solid or liquid products.

Direct heating is also used for bath concentration. It is even the oldest application with the concentration of sulfuric acid in the chemical industry.

III INDIRECT HEATING OF LIQUIDS (TIC - EIC) :

III.1 Compact Immersion Tube : (Fig. 4)

III.1.a Description :

The compact immersion tube consists of three parts : the burner, the combustion chamber and the exchanger tube.

The flame is developed in the combustion chamber which must be as compact as possible. Thus, the burner must maintain intense combustion which must be complete with only slight excess air.

The combustion chamber is cylindrical in shape with a cone at one end for hook-up to the exchanger tube.

There are several possible configurations for combustion chamber installation. One solution is to install the Combustion chamber outside the tank which contains the liquid to be heated. In this way, the combustion chamber is cooled by the air drawn in for combustion or it is insulated by refractory material. It is also possible to place the combustion chamber in the liquid to be heated. It can be mounted either vertically or horizontally.

By increasing gas flow velocity, the small diameter exchanger tube makes possible an increase in heat transfer and consequently an improvement in heating efficiency.

This reduction in diameter also noticeably increases head loss in the tube. The implementation is the result of a compromise between efficiency gains and permissible air and gas supply pressure.

III.1.b Performance data : (Fig. 5)

Generally speaking, the compact immersion tubes are designed to provide heating efficiency of between 92 % and 94 % on NCV, for the following conditions : temperature of liquid heated 70 °C, excess air in burner, 20%.

Because of its simplicity of operation and its low cost, this type of set-up is particularly well suited to installations which require a capacity of below 150 KW on NCV or for special dimension tanks (very long or shallow).

III.2 Compact Immersion Exchanger : (Fig. 6)

III.2.a Description :

The main characteristics of equipment is :

- *Compactness : High combustion product heat transfer is obtained by producing a turbulent flow along the heat exchanger walls without increasing the heat exchange surface area.*

IMMERSION EXCHANGER	CAPACITY (KW on NCV) 100	200	300	400	500	DIMENSION (m) l x L x H
EIC 250	——	——				0.15 x 1.3 x 0.5
EIC 450		——	——	——		0.25 x 1.8 x 0.5

- *High efficiency : This simple technology which implies improved reliability results in efficiencies greater that 92 % on n.c.v.*

The compact immersion heater is roughly a rectangular parallelepipede comprising two parts : the combustion chamber directly connected to a premixed burner and containing a number of opened vertical tubes through which the liquid to be heated flows. The products of combustion leave the combustion chamber after having heated the liquid which moves from bottom to top by natural circulation. The second heat exchanger stage is designed so that combustion products follow available paths in the exchanger walls.

III.2.b Combustion Product Heat Transfer

The transfer of combustion product heat increases due to local changes in velocity and thus in kinetic energy. Theses changes occur when the flue gases impinge on the baffle plates or waffled plates of the exchanger walls.

III.2.c Performance data : (Fig. 7)

The study of various alternatives of heating a liquid to 70 °C for instance has shown that :

- *heat exchangers using a heating fluid from central boiler rooms have a high output per unit area but a relatively low overall efficiency not exceeding 70 % ;*

- conventional immersion tube heat exchangers have a low output per unit area but a higher efficiency of 70 to 80 % on n.c.v. ;
- compact tube heat exchangers may have efficiencies of 90 % on n.c.v.
- compact immersion heaters have a high output per unit area and an excellent efficiency of 92 to 96 % on n.c.v.

III.3 Applications :

III.3.a Surface treating :

. Galvanizing : Degreasing baths and fluxing.
. Metal plating : Degreasing baths, phosphate coating, acids, copper plating, chrome plating, burnishing, filling-up, etc.
. Enameling : Degreasing baths, neutralization.
. Painting : phosphatizing baths, passivation.

III.3.b Agriculture and food processing :

. Food processing : Hog scalding vessels, steeping, bleaching, hot water production.
. Agriculture in greenhouses : hydroponics, hot water supply by radiant systems.

III.3.c Textile Industry :

. Textile : dyebaths and bleaches, washing, hot water production.

IV CONCLUSION ABOUT HEATING OF LIQUIDS :

Various installation configurations are possible, depending upon the position of the heating equipment with respect to the liquid and upon temperature distribution. There are three installation configurations : the "in-situ" configuration, the shunt configuration and the hot water generator configuration. The heating techniques are included as examples.

IV.1 "In Situ" Set-up :

The heating unit is located in the industrial tank (or machine) itself. The available room in the tank is sufficient to mount the unit. For this type of set-up, the bath, generally, is not under pressure (the free surface is under atmospheric pressure) and its temperature is practically uniform.

The heating equipment increases and holds the bath temperature constant.

This is the configuration of maximum efficiency.

IV.2 Shunt set-up :

When there is not sufficient available space in the industrial tank to install a heating unit, a circulation loop is installed outside the bath, between the industrial tank and the heating unit. To hold heat loss to a minimum, efforts will be made to install the tank and the heating unit as close as possible to each other. The heating unit inlet/outlet temperature gradient can vary from several degrees to several tens of degrees depending upon the type of set-up.

When the industrial liquid is under pressure, the heating unit is a boiling device.

When the industrial tank is at atmospheric pressure, the heating unit is an adjoining tank in which the heating equipment (submerged combustion, compact immersion tube or exchanger) is installed.

With this type of set-up, the heating efficiency of the liquid reflects allowances for losses to exhaust gases, losses at the walls of the heating unit and distribution losses. The latter two types of losses are on the order of 1 to 3%.

IV.3 Hot water generator set-up : (Fig. 8)

The generator and the network which makes use of the hot water form an open circulation loop. Usually, the hot water, once used, is not recycled. Thus, the generator is fed cold water and produces hot water continuously. The cold water can be preheated via an exchanger by the spent hot water before is flows out the drain.

For maximum heating efficiency, products of combustion flow countercurrent to the water which is heated. Before being exhausted, the flue gases are cooled by coming into contact with the coldest water. The hot water generator can therefore be combined with a storage unit to offset fluctuations in hot water consumption.

Various types of equipment can be used as hot water generators. Among them :

- the submerged combustion hot water generator consists of a storage tank in which the heating equipment is installed. For a water temperature of less than 60 °C, heating efficiency exceeds 100 % on NCV. Beyond 60 °C, the submerged combustion tank is equipped with a fume tower scrubber which is supplied with cold water.

- the hot water generator with an immersion exchanger is installed in a small storage tank. At hot water temperatures below 90 °C, heating efficiency exceeds 100 % on NCV.

V OTHER BURNERS FOR DRYING :

The development of modern techniques in gas utilization often calls for burners producing a flame for a very specific purpose. The burner is not only a source of heat, but also a tool with a specific function.

V.1 Catalytic Combustion Radiant Burners : (Fig. 9)

These systems induce a catalytic oxydation of fuel with air without any flame. This is done through contact with porous mass impregnated by a catalyst, generally being platinum-based.

The schematic diagram of an example of a system operating with partial premixture of air and gas and forced convection is shown in figure 9.

Catalytic systems are interesting in that they can be utilized in explosive atmospheres: as long as there are no ignitable materials, such as rags for example, most industrial solvent fumes when in contact with the porous mass, even when brought up to nearly 1000 °C, do not provoke ignition, but only a catalytic oxidation reaction.

On the other hand, the emission spectrum of these burners corresponds to the absorption spectrum of the most commonly used paints. Therefore, these systems are very appropriate for drying paint.

V.2 Air Duct Bar Burners : (Fig. 10)

The purity of the common high CV gaseous fuels, especially the absence of sulfur, permits the products of combustion to be used for the direct heating of air for use in many heating processes, such as drying.

To do this, a special type of burner has been developed called "air duct" bar burner because it is placed directly into ventilation ducts.

This type of burner and its location in a ventilation duct is illustrated in figure 10.

The temperature of the hot gases obtained can reach 400 °C.

The utilization of the hot gases resulting from the dilution of the combustion products is common in the area of drying.

Some air duct burners are designed to provide incineration of combustible gaseous effluents, provided that the oxygen percentage in the effluent is 18% or more and that the dilution air temperature is less than 540°C. This incineration takes place at a temperature of 800 to 850°C.

VI CRITERIA FOR CHOOSING INDUSTRIAL SYSTEMS :

Once the nature of a heating operation has been definitely established, the choice of the heating system to use can be made by first considering criteria of a general nature, relating to the burner's operation, independently from its specific application, and then by criteria of a specific nature concerning the product to heat the furnace, if it already exists.

VI.1 General Criteria :

These criteria concerning the operation of the burners used must be considered in order to fulfil the operation requirements.

The temperature required is of great important to the limits of the temperature ranges available: low temperatures will often call for burners operating on excess air or on recycled combustion products ; at high temperatures, preheating and/or over oxygenation of the combustion air will be useful or necessary.

The nature of combustion products required, outside of any other consideration, can determine by itself the choice of the burner : air deficiency combustion with complete oxygen utilization, excess air combustion without unburned products, absence of NO_x.

The turn down ratio required, which is the ratio of maximum to minimum burner capacity at its nominal setting, is a criterion of selection related to the burner's fuel supply and regulation system.

The turn down ratio required of a system may be obtained by output modulation of the burner or burners, by sucessive shut down of some burners, or by a combination of these procedures.

It is only after having examined these different possibilites with regard to process requirements and the heating techniques applicable, that the flexibility of each burner can be determined.

Adjustment flexibility, which depends on the aeration limits within which the burner still operates normally, is an important factor for multi-usage equipment when one wishes to put in operation simplified control systems or when system variations influencing the equipment settings are important.

The noise produced by the burner becomes a selection criterion to be taken into account in certain instances. It is important to note that it is not an absolute necessity but a criterion related to the burner/furnace connection, in some cases.

There is really very little relation between the noise produced by a burner tested in open air and that produced by the same burner in a furnace. The noisiest burners in open air prove to be, in some cases, quieter than others when operating in a closed space.

VI.2 Criteria of a Specific Nature :

These criteria deal with the product to be heated and the oven designated for this purpose, provided that the design or construction of the latter has already been made at the time of the burner selection.

The thermal receptivity of the product to be heated could be defined as its capacity to absorb the produced heat directly or indirectly by the flame.

It is not a physical parameter, but a concept covering what common sense designates as heating capacity and is the result of a combination of physical variables concerning the product to be heated.

The thermal receptivity of the product to be heated depends, among other things, upon :

- its thermal conductibility, that is, its capacity to diffuse toward the inside the heat received on the surface,
- its radiant absorption coefficient, that is, its capacity to transfer into heat a fraction of the radiation hitting the surface,
- its specific surface, that is, the surface available for receiving heat through convection or radiation per mass unit to be heated,

- its oxidation capability; that is, its capacity to develop surface films or dross reducing heat transfer.

The value of these parameters determines some heating processes.

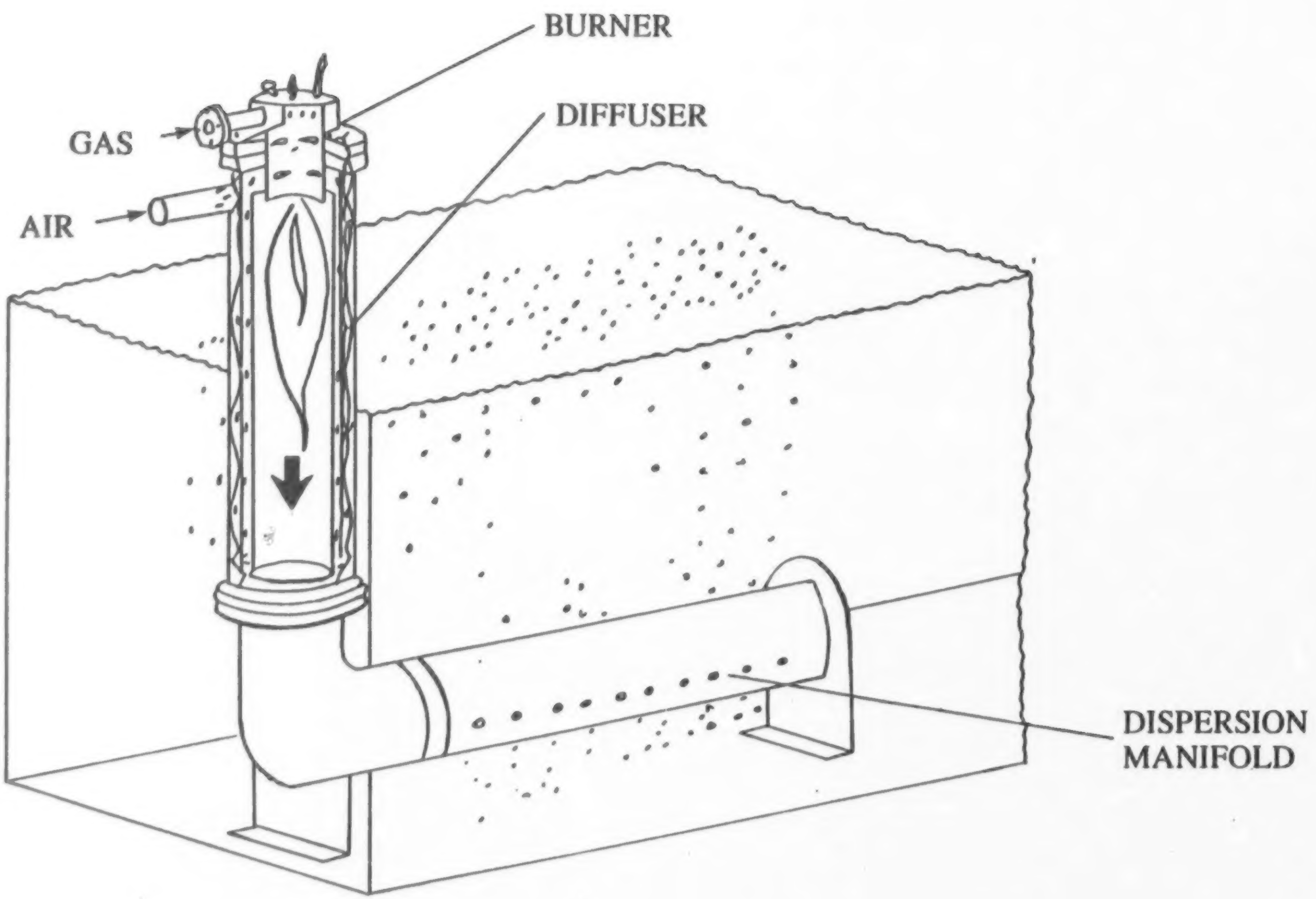
The furnace receptivity corresponds to its capacity for being an efficient heat transfer medium between the flame and the product to be heated.

The thermal receptivity of the furnace depends, among other things, upon :

- its thermal insulation, that is, its capacity to avoid heat loss through the walls,
- its thermal inertia; that is, its capacity for storing heat. This capacity is not conducive to good heating efficiency during the temperature build up period, but contributes to the stability of the operation.
- The presence of walls : crucibles, blocks, semi-blocks, radiant tubes, being obstacles to heat transfer between the flame and the product to be heated,
- the design of the combustion products outlet, providing (or not, or badly) preheating for the product to be heated,
- the presence of heat recuperators for combustion air preheating.

VII.3 Applications :

- Buildings of average height, poorly insulated, poorly closed
- Heating of stations or areas
- Intermittent heating.
- Installation height 3.50 to 8 metres from the ground.


BURNER
DIFFUSER
GAS
AIR
DISPERSION MANIFOLD

SUBMERGED COMBUSTION EQUIPMENT

FIGURE 1

FIGURE 2

SUBMERGED COMBUSTION EFFICIENCY
DIAGRAM FOR WATER

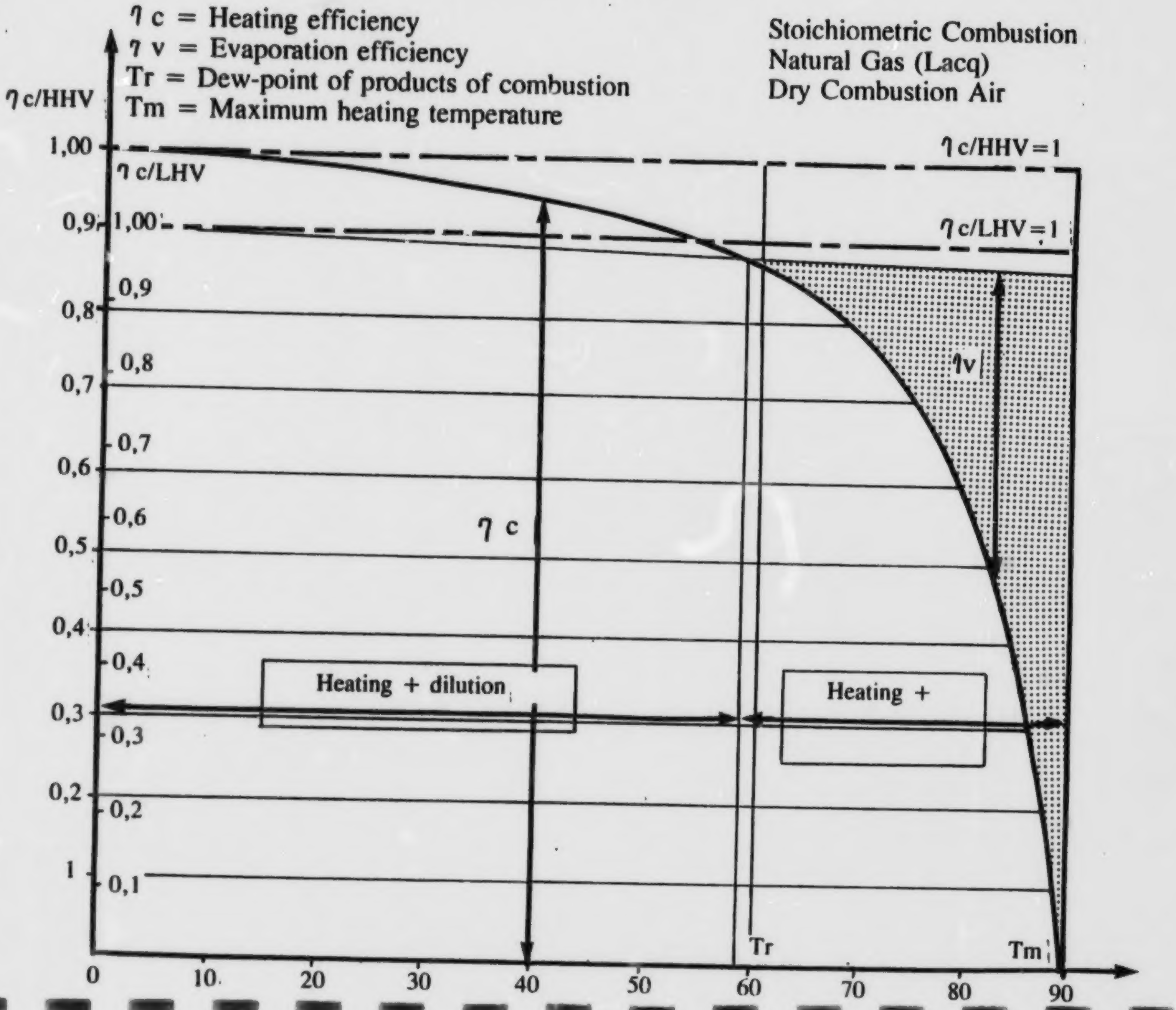


FIGURE 3

SUBMERGED COMBUSTION APPLIED TO WATER

Variation in heating efficiency with temperature
and excess combustion air

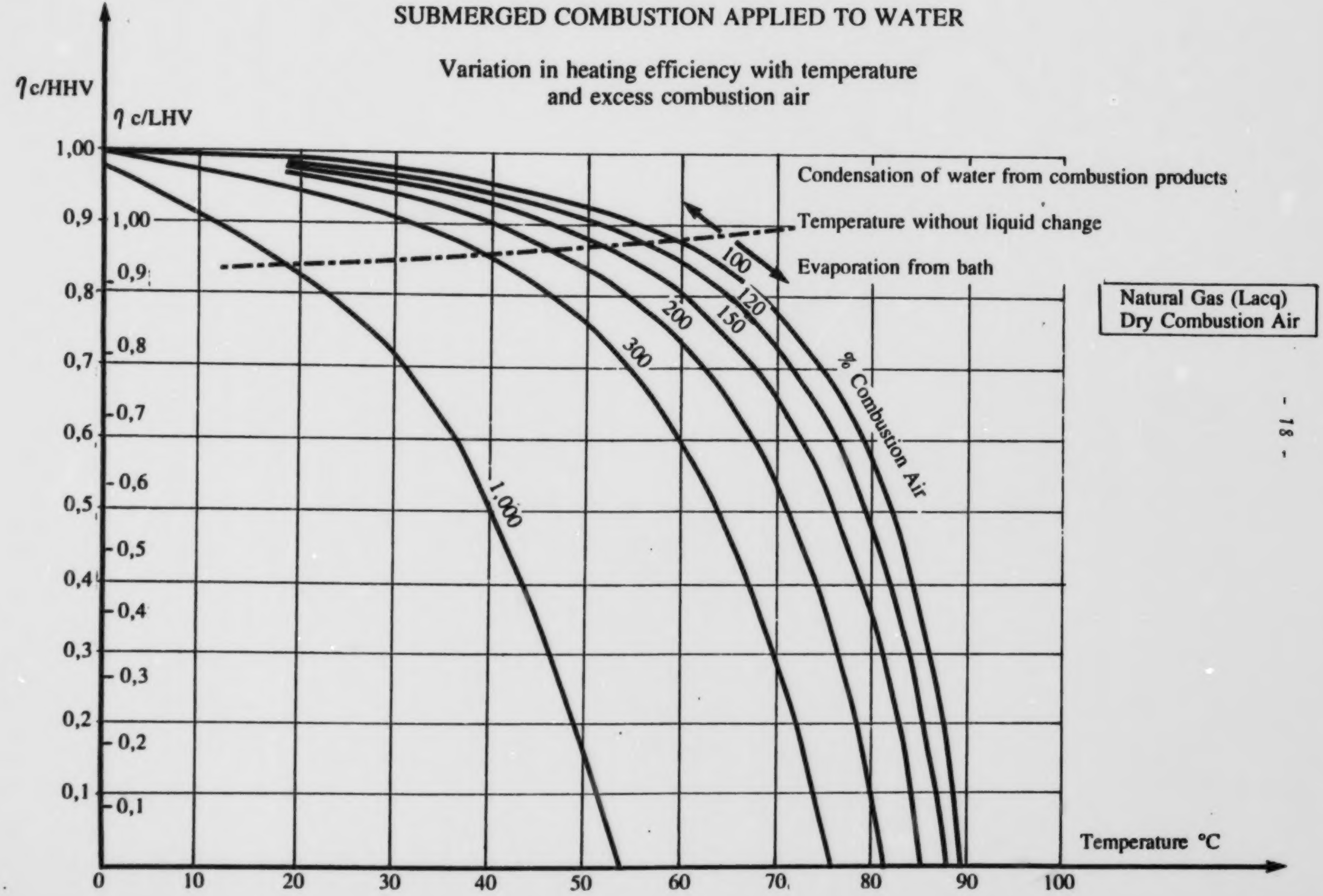


FIGURE 4

COMPACT IMMERSION TUBE

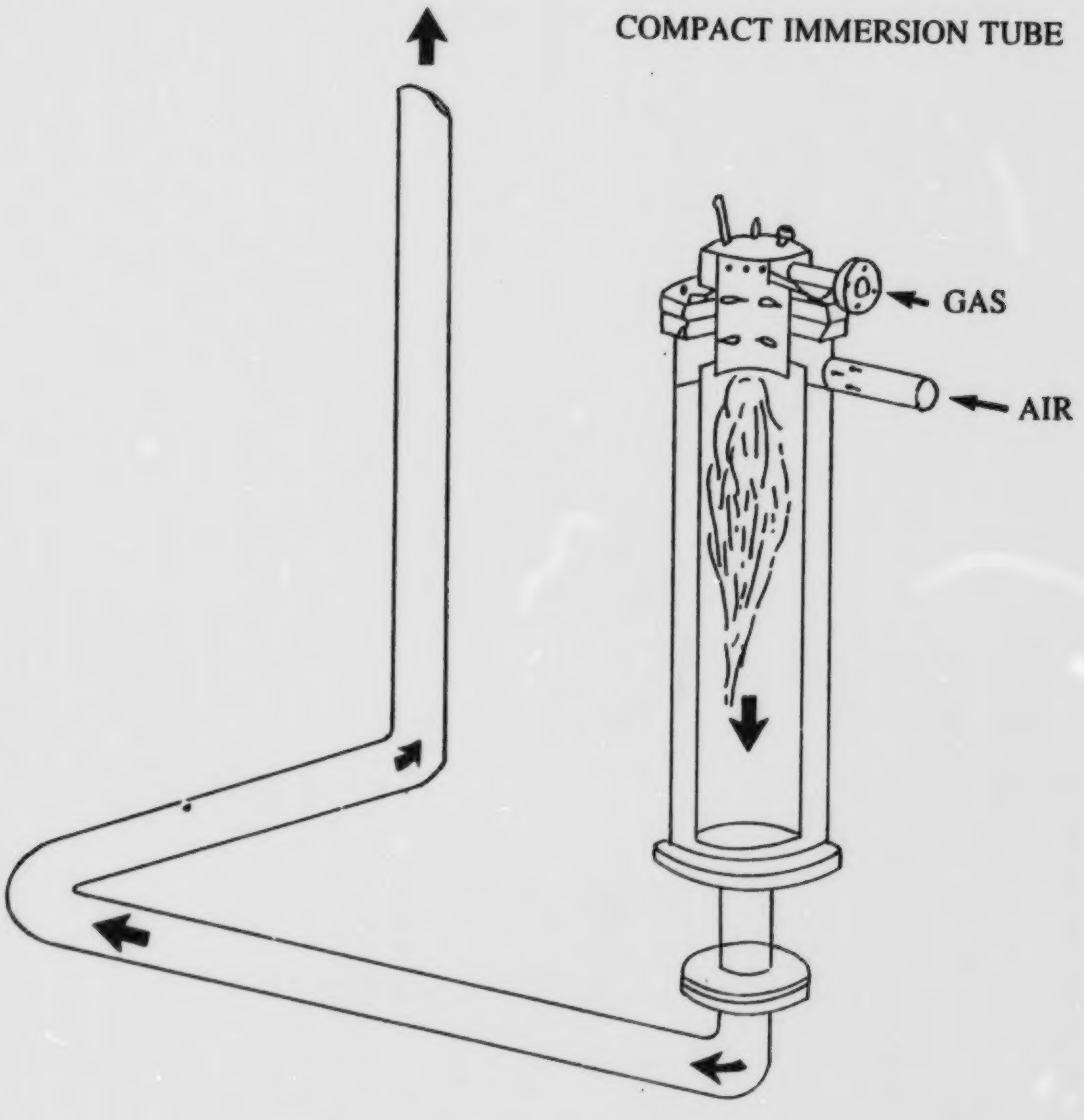
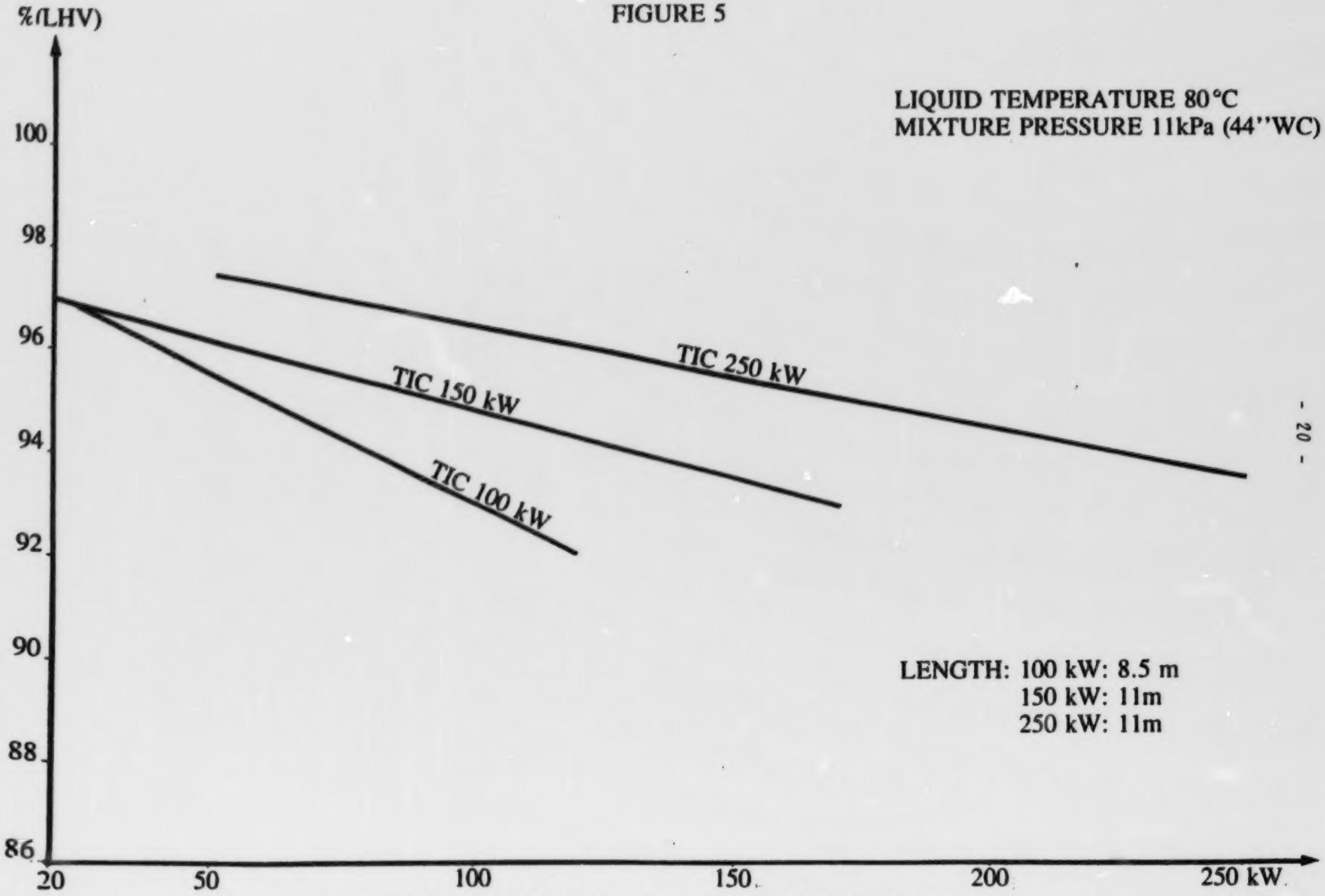
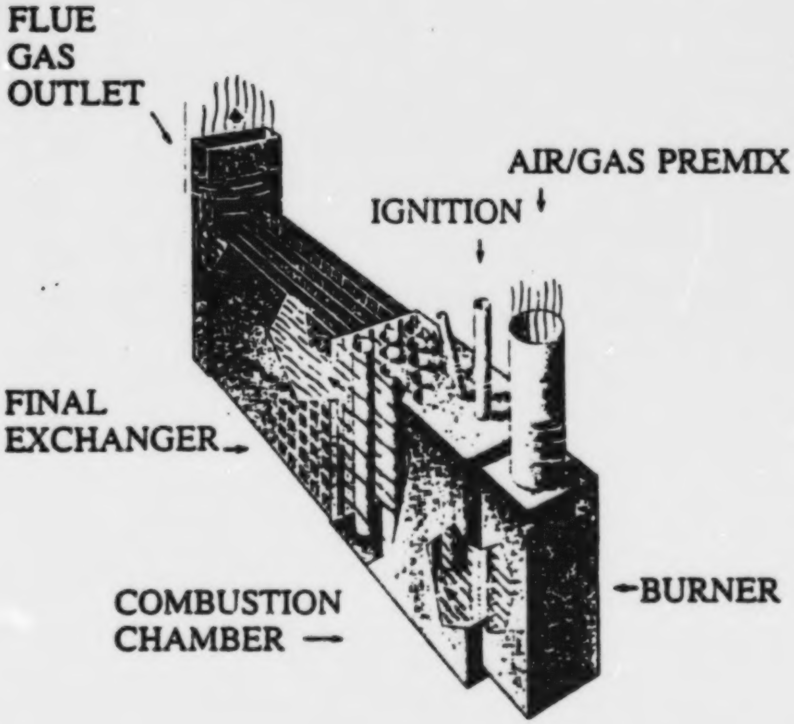


FIGURE 5



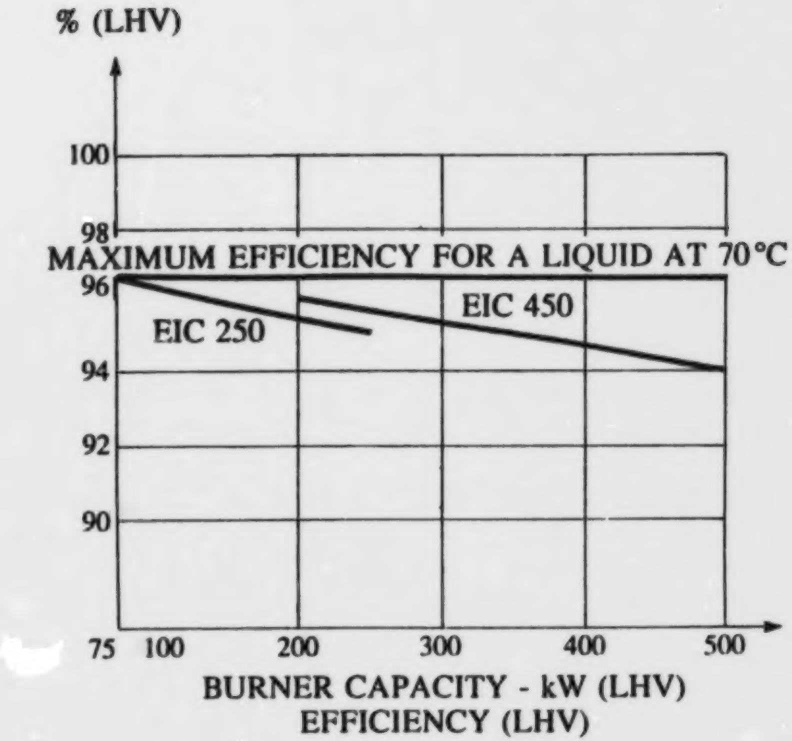
EFFICIENCY (LHV) OF COMPACT IMMERSION TUBES

FIGURE 6


FLUE
GAS
OUTLET
AIR/GAS PREMIX
IGNITION
FINAL
EXCHANGER
BURNER
COMBUSTION
CHAMBER

COMPACT IMMERSION EXCHANGER

FIGURE 7



EFFICIENCY OF COMPACT IMMERSION EXCHANGERS

FIGURE 8

HOT WATER GENERATOR ARRANGEMENT

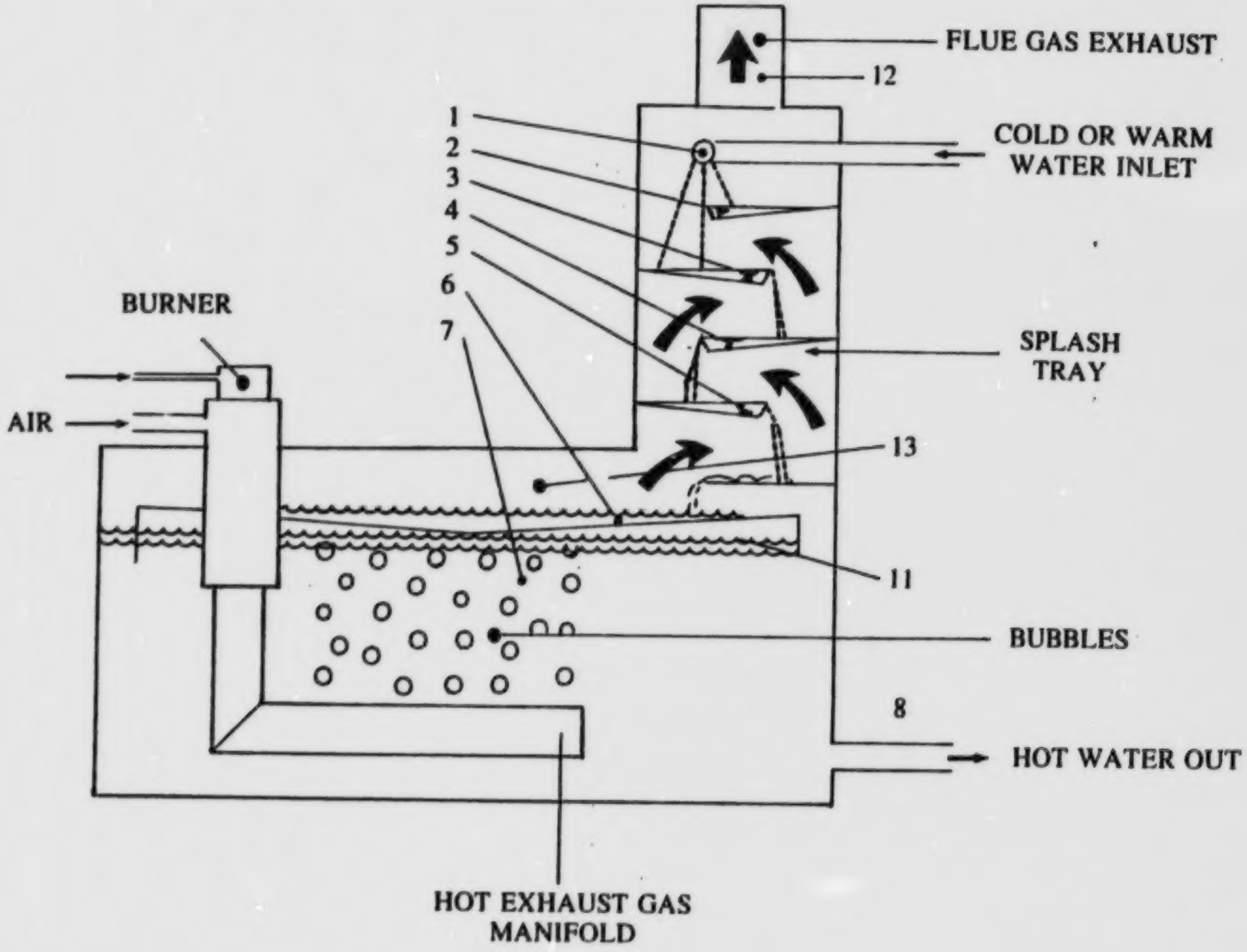

FLUE GAS EXHAUST
12
1
2
3
4
5
6
7
COLD OR WARM WATER INLET
BURNER
AIR
SPLASH TRAY
13
11
BUBBLES
8
HOT WATER OUT
HOT EXHAUST GAS MANIFOLD

FIGURE 9

CATALYTIC RADIANT BURNER
GAS

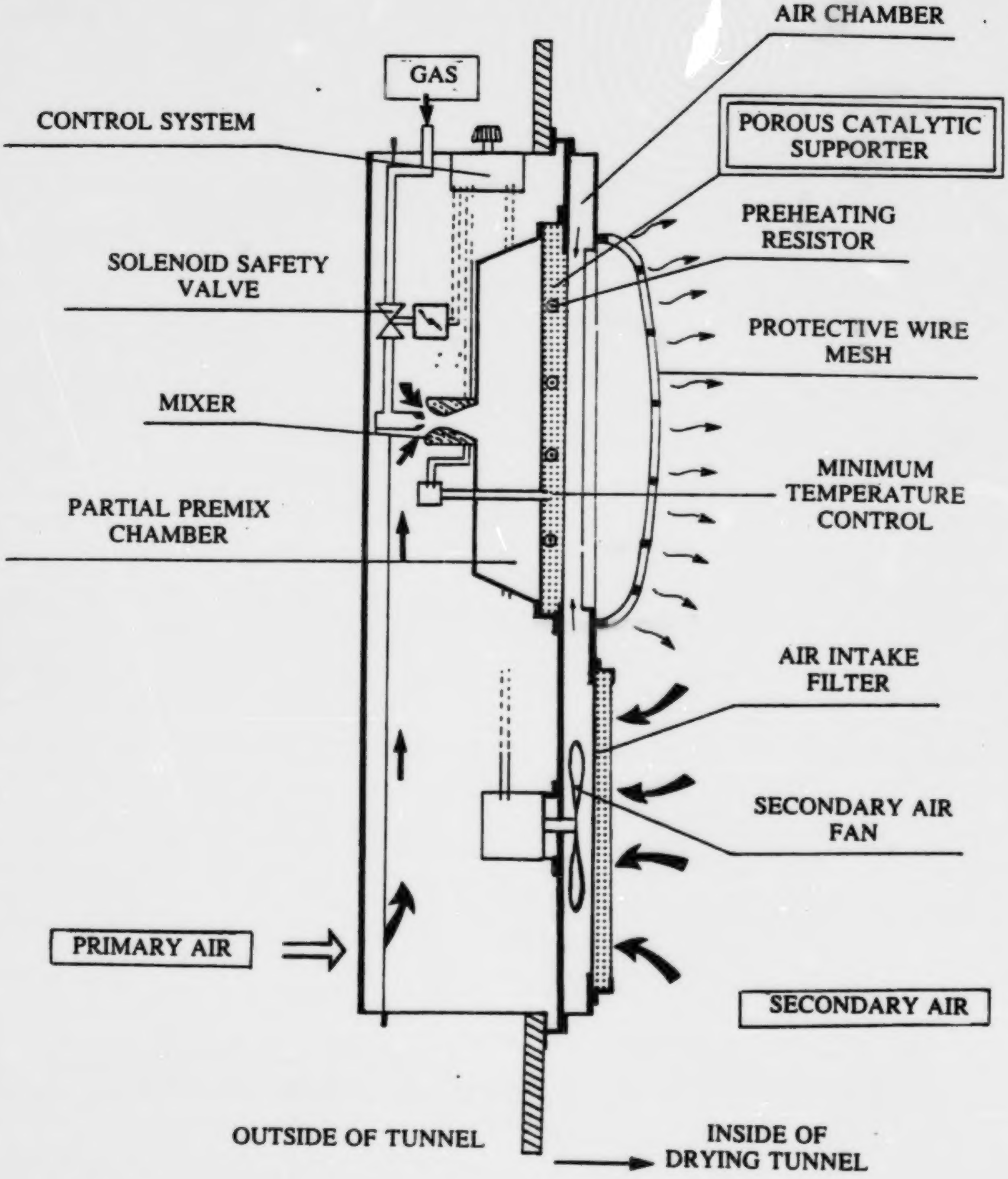
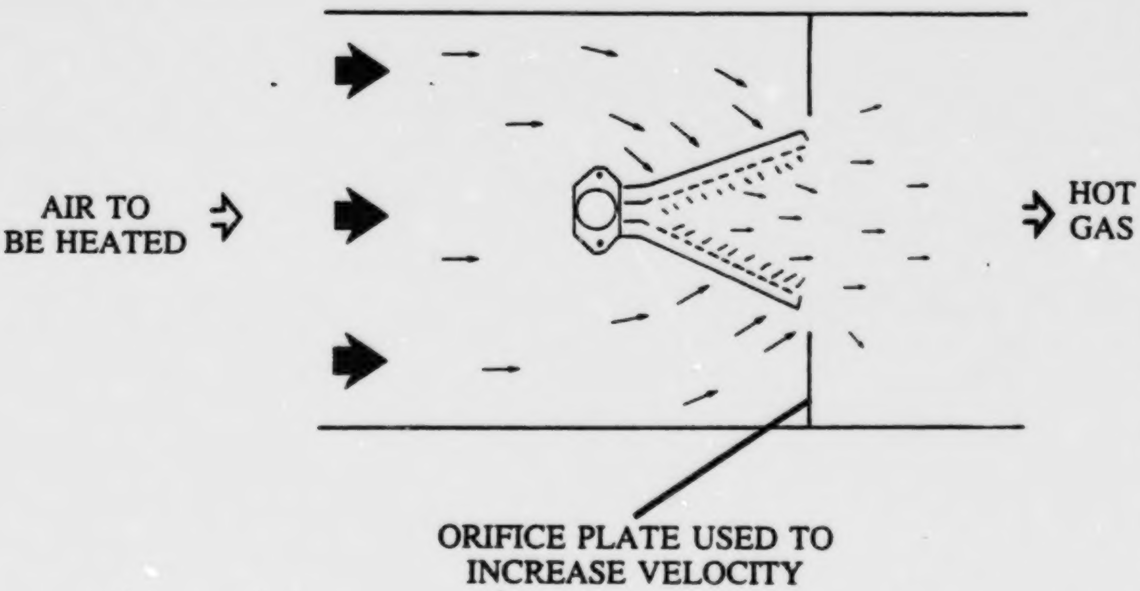
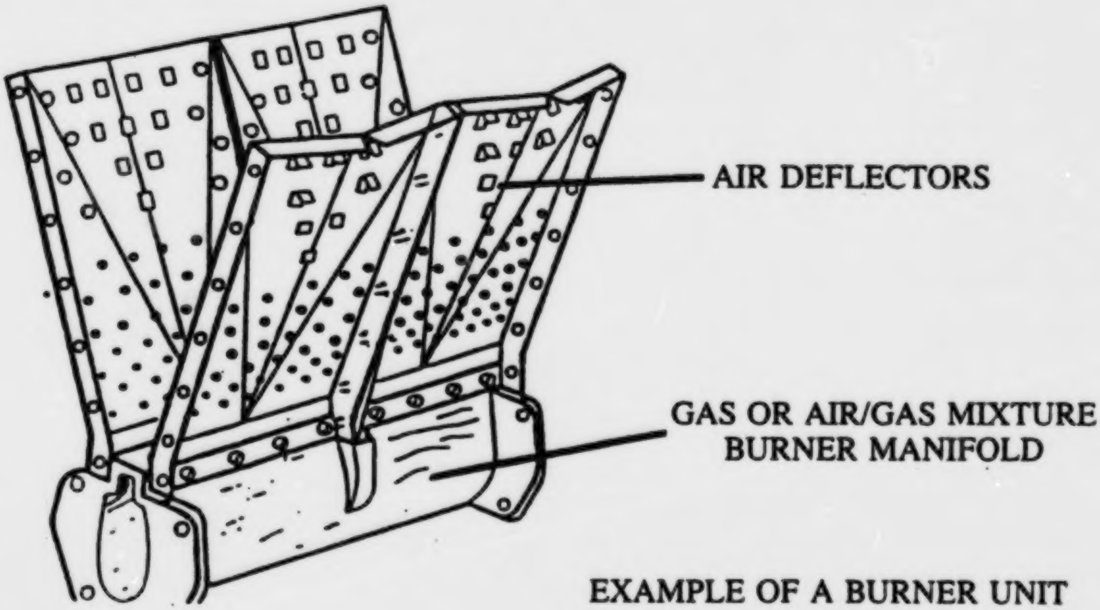

AIR CHAMBER
GAS
CONTROL SYSTEM
POROUS CATALYTIC SUPPORTER
PREHEATING RESISTOR
SOLENOID SAFETY VALVE
PROTECTIVE WIRE MESH
MIXER
MINIMUM TEMPERATURE CONTROL
PARTIAL PREMIX CHAMBER
AIR INTAKE FILTER
SECONDARY AIR FAN
PRIMARY AIR
SECONDARY AIR
OUTSIDE OF TUNNEL
INSIDE OF DRYING TUNNEL

FIGURE 10

AIR HEAT BURNER
FOR IN-DUCT MOUNTING


AIR TO
BE HEATED
HOT
GAS
ORIFICE PLATE USED TO
INCREASE VELOCITY


AIR DEFLECTORS
GAS OR AIR/GAS MIXTURE
BURNER MANIFOLD

EXAMPLE OF A BURNER UNIT

SECTION

VII

NEW TECHNOLOGIES TO INCREASE COMBUSTION EFFICIENCY AND REDUCE NOx EMISSION

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Normand Brais
Todd Combustion Limited

9/5/666

ABOUT THE SPEAKER:

Normand Brais, Technical Director of Todd Combustion Ltd., is a graduate from Ecole Polytechnique de Montreal. Since 1982, he has been professor of "Fundamental and Applied Combustion" at the Graduate and Continuing Education Faculty of the University of Montreal.

Mr. Brais joined Todd Combustion in 1983 as Chief Engineer of the analog modelling laboratory for fluid flow simulations. In 1984, he was given the direction of the engineering department and the research and development activities.

The majority of Mr. Brais' career has been concerned with analog simulation of jets and fluid flows, low excess air combustion, pollutants formation, pulverized coal combustion, rotary kiln burners and mathematical modelling of flames. He has published several papers on these subjects. He is also a member of a combustion expert committee for the NSERC of the Canadian National Research Council.

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ABSTRACT

To meet tomorrow's environmental challenge, combustion equipment designers must seek for new methods and approach the problems in a more comprehensive manner. The semi-empirical methods actually in use have long reached their limits as far as the flame structure prediction is concerned.

A new technology which uses mathematical models applied on a discrete grid to simulate a flame in its environment is presented here. The flame structure is thus predicted and a two-step approach is used to model the formation of thermal NOx. The first step involves the main exothermal reation of methane in air, and the second involves the solution of the Zeldovich reaction scheme for the generation of nitrogen oxides.

A single step reaction scheme is used for the methane combustion calculation which is kinetically controlled. The limiting control of turbulence is taken into account by using a reaction rate which is the minimum to that given by the Arrhenius and Eddy-Break-Up models. A flux model is used to calculate radiative heat transfer.

In the second step, a chemical equilibrium analysis is performed to determine the concentration of radicals (O, OH, and H) in the product stream and conservation equations are then solved for N and NO which utilise the convective and diffusive transport terms calculated in the first step.

The method is applied to a large industrial burner featuring an axial and a swirling recirculating flow region into which fuel is injected either tangentially or axially. Parametric variations on the burner geometry and operational variables are then studied in terms of their impact on NOx emission.

CONTENTS

I- Introduction

II- NOx reduction methods

III- Description of the model

IV- Application to burner design

V- Discussion of results

VI- Conclusion

- Nomenclature
- References

I- INTRODUCTION

Combustion generated pollution is now recognised as a threat to the environment. The most obvious air pollutant is smoke, being visible to the naked eye. Other pollutants of importance are, carbon monoxide (CO), unburned hydrocarbons, sulfur oxides (SOx) and finally nitrogen oxides (NOx). Virtually all of these pollutants have a harmful effect on the ecosystem [1]. Emotive phrases have evolved over the years which described these effects , as in the "greenhouse effect", "acid rain", "city smog", and the "ozone layer hole".

In this work we concentrate on the last pollutant mentioned, NOx which is responsible amongst other things for the apparent depletion of the earth's ozone layer through a chain reaction mechanism. The quantitative influence of this chain mechanism in the stratosphere is uncertain. However, this gap in knowledge and the fact that a hole in the ozone layer above the antarctic has increased alarmingly in size in recent years, has led to a worldwide tightening of legislation governing the sources of NOx emission.

The term NOx usually implies two major oxides, nitrogen oxide (NO) and nitrogen dioxide (NO2). In combustion , NO is the dominant of the two components, NO2 being mainly derived from NO. It is the prediction of NO we are concerned with in this paper, loosely referred to as NOx. The following passages lead to NO production.

- From reaction of N_2 with oxygen, "thermal NO",
- from N existing in the fuel, "fuel NO", and,
- from reaction of fuel derived radicals with N_2 ultimately leading to NO, the "prompt NO".

It is the first of these three mechanisms we are concerned with here, since it is the one that can be most affected by burner design.

Since, air borne pollution is no observer of human boundaries, world wide concern over the issue has come under the UN/ECE convention on "Long Range Trans-Boundary Air Pollution", which is negotiating an international standard on NOx emissions. Similar activities have been undertaken in the EEC, where particularly stiff controls already exist in power stations operating in Germany and the Netherlands. Typically, Dutch legislation requires limits of 200 ppm NOx for liquid fuel and 120ppm NOx for gas fuel industrial burners [3]. In the near future these figures are expected to be halved.

It is under this climate of tightening legislation that new improved methods of design are called for to replace existing burners/combustors, and improved methods of operation required to prolong the use of existing units.

III- DESRIPTION OF THE MODEL

The problem of simulating a real burner accurately still remains a formidable one. Not only has one to contend with complex flow fields which often include swirl and regions of flow reversal and are highly turbulent, but also one has to tackle complex chemical kinetics which govern the combustion and subsequent formation of trace chemicals.

It is clear that the combustion and NOx reactions in a burner are inseparable, and the latter is a direct consequence of the former. However, it has been observed in experiments [1] that the main oxidising reaction between fuel and aire is not affected by the formation of trace species which lead to NOx. Hence, although the NOx reactions tend to be endothermic, the radicals involved occur in minute concentrations and they do not affect the exothermic reaction occurring in the bulk of the burner. On the other hand, as pointed out earlier, the temperatures generated by the combustion are the determining factor in NOx formation. With these observations in mind, one can treat the combustion and NOx formation steps as separate.

Conservation Equations Solved

Flow and reaction in a burner is defined in terms of a set of elliptic partial differential equations that express the conservation of mass, momentum, energy and other fluid variables in three-dimensional recirculating flow. With the primitive variable representing velocity, chemical species, temperature and turbulence variables the equations can be written in a generalised form:

$$\frac{\partial}{\partial t}(\rho\varphi) + \mathrm{div}(\rho\vec{V}\varphi - \Gamma_{\varphi}\,\mathrm{grad}\,\varphi) = S_{\varphi} \qquad (1)$$

where $\rho, V, \Gamma_{\varphi}$, and S_{φ} are density, velocity vector, turbulent exchange coefficient and source rate per unit volume respectively. The sources and exchange coefficients for velocities and turbulence have been discussed in detail elsewhere [12, 13] together with the effects of buoyancy. The K-ε model of turbulence is used to calculate the effective viscosity [13] in all regions of the flow, except where strong swirl exists. In these regions a mixing length model is used instead.

The numerical analogue of equations (1) was solved using PHOENICS on a cylindrical polar finite difference grid.

II- NOx REDUCTION METHODS

The main factor controlling thermal NOx is temperature. In fact, NOx formation is insignificant at temperature below 1800K. In has been observed [4] that:

$$NOx \propto e^{.009T}$$

where T=reaction temperature in Kelvin. All other factors can be viewed as means of affecting temperature. Therefore, the burner designer needs to avoid hot spots and also keep the time available for NOx formation to a minimum. Both these factors are affected by an increase in the flow of air into the combustion zone. However, care needs to be exercised if CO and unburned hydrocarbon levels are to be kept to a minimum. Practical means of control include:

- Low excess air combustion since O_2 concentration is reduced.
- Exhaust, or flue gas recirculation effective if the recirculation gas is cooled before being injected into the combustion zone.
- Water injection into the primary combustion zone of a burner as used in industrial turbines; utilizes the latent heat of water droplets to reduce temperature.
- Staged combustion, which introduces primary, secondary or even tertiary combustion zones and hence allows tighter control on temperature limits. In order to be effective, both air and fuel delivery needs to be staged. Desing complexities result.

In order to be able to evaluate the influence of these factors, a mathematical model which represents correctly the physico-chemical aspects of the problem is needed. The object of this paper is to report the development of such a model and present preliminary calculations.

Combustion Model

Calculations were performed using a single step combustion model which could be either diffusion controlled or kinetically controlled. In both cases the equation for a mixture fraction [14] is solved; in the first case the mass fraction of fuel at any point is determined from it, while in the second, an additional conservation equation is solved for the mass fraction of fuel. The mass fractions of products and oxidant are then derived from these quantities [15].

The disappearance rate of fuel (CH4) was calculated in the kinetically controlled model as being the minimum of that given by an Arrhenius expression and eddy-break-up expression [16].

$$R_{fu} = -C_R \rho \frac{\varepsilon}{k} \min\left[m_{fu}, \frac{m_{ox}}{s}, \frac{m_{pr}}{1+s}\right] \tag{2}$$

where, C_R either takes a constant value of 4 or is calculated locally as a function of k and ε ie.

$$C_R = 23.6\left[\frac{\mu}{\rho}\cdot\frac{\varepsilon}{k^2}\right]^{0.25} \tag{3}$$

The effect of radiation was included using a flux model [15, 17] which solves ordinary differential equations for radiation fluxes along each coordinate direction.

The ideal gas law was used to calculate density variations and the specific heat was allowed to vary as a function of temperature and composition.

The combustion calculation yielded field values of all the dependent variables, which were stored on disc for use in the NOx prediction.

NOx Calculations

The chemistry involved in the formation of NOx will now be described. The model employed incorporates one of the simplest and most widely used mechanisms for calculations involved nitric oxide formation, namely the Zeldovich mechanism:

$$N_2 + O = NO + N , \qquad (4)$$

$$O_2 + N = NO + O , \qquad (5)$$

$$OH + N = NO + H , \qquad (6)$$

As mentioned earlier, the basic assumption in this work is that reaction (4)-(6) can be considered as decoupled from the main fuel-burning reaction. In addition, the species appearing in these reactions are assumed to form part of the product stream of the single combustion reaction which in the first step of the calculation was assumed to be a single entity.

The species O, O_2 , OH, H, H_2 , CO and CO_2 are assumed to exist as chemical equilibrium. Their concentration is determined from a minimisation of Gibbs free energy subject to the following constraints: (a) mass balance of the elements present in the system. (b) specified enthalpy and, (c) specified pressure. The polynominal equations for species concentrations are solved in a single pass by calling the code CREK [18] in the ground station of PHOENICS. The mass fractions of N and NO are determined by solving conservation equations (equation (1)) which have as convection and diffusion fluxes the ones determined in the combustion step. It is assumed that the concentrations of the other species remain invariant, with the justification that they are produced by reactions which are considerably faster than those involving nitrogen.

The source term for the two conservation equations comes from a summation of the reaction rate for all reactions in which each species participates; ie. the Zeldovich reactions. The forward and backward reactions are given by Arrhenius expressions of the form:

$$R = K\rho^2 F_1 F_2 \qquad (7)$$

where F_1 and F_2 are the concentrations of participating species and K the reaction rate coefficient, given by

$$K = 10^B T^N e^{-T_{act}/T} \tag{8}$$

with $10^B T^N$ being the pre-exponential factor, and T_{act} the activation temperature. The values of these quantities are given in Table 1.

REACTION	FORWARD			REVERSE		
	10^B	Tact	N	10^B	Tact	N
$N_2 + O = NO + N$	7.6×10^{10}	3.8×10^4	0	1.6×10^{10}	0.0	0
$O_2 + N = NO + O$	6.4×10^6	3.15×10^3	1	1.5×10^6	1.95x10	1
$OH + N = NO + H$	6.3×10^8	0.0		2.5×10^9	2.45x10	0.5

Table 1: Constants used for the forward and reverse reactions (Equations 7, 8)

IV- APPLICATION TO BURNER DESIGN

The above described model was derived to predict the NOx emissions of a typical burner used in large steam generators.

It widely recognised [19], [20] that significant NOx emission reduction can be achieved through burner design. However, better understanding of the flame structure produced by a given burner is necessary in order to identify and apply successful design modifications.

Such modifications involve fuel injection distribution and pressure, air flow characteristics, and swirl stabilisation. Furnace geometry also needs to be considered since it conditions the heat transfer rate and the combustion products flow pattern [21].

The burner under investigation is the Todd axial air flow type using a blade swirler to impart a tangential velocity component to a portion (15%) of the total air flow. The latter principle is commonly used to stabilize high intensity industrial flame [22].

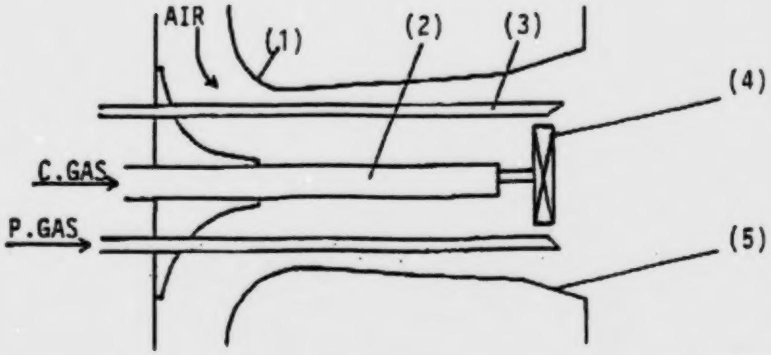


Figure 6: Typical Burner Investigated

As shown in Figure 6, the burner major components include a venturi air register (1), a central fuel injector (2), six peripheral fuel injectors (3), a swirler (4), and an exit quarl (5).

The characteristics of the burner being simulated were:

- Maximum heat output: 100 MMBTU/hr
- Combustion air temperature: 500 degrees F.
- Burner air side pressure drop: 10" WG
- Fuel: Natural gas (100% CH4)

A grid was defined using a polar coordinate referential which simulates a cylindrical combustion chamber 9 ft. in diameter by 20 ft. long.

At the upstream end of this combustion chamber, inlet boundary conditions were set, thus allowing for the burner outflow to be specified. A constant pressure boundary was assumed downstream, at the exit.

The cylindrical combustion chamber was selected initially to reduce computing cost. However, in the course of our parametric study, rectangular furnaces were also simulated.

This parametric study covered the influence of furnace size, combustion air temperature, fuel injection mode, burner pressure drop, burner heat input, and number of burners on thermal NOx emission tendencies. Some highlights of the most significant results are presented here.

4.1 Central Fuel Injetion Mode

4.1.1 Influence of air temperature on flame structure and NOx emissions

Results are presented in Figures 7 to 9 where the combustion air temperature is increased from 70 degrees F. to 300 degrees F. and finally to 500 degrees F. The flame temperature contours reveal the evidence that flame length is decreasing as combustion air temperature increases. NOx concentration contours in the flame are shown in Figures 10 to 12 and display a significant increase when air temperature is increased. A synopsis of these results is presented in Table 2.

COMBUSTION AIR TEMP. (F.)	70	300	500
Primary Air ratio %	15.5	13.3	12.03
Adiabatic Flame temp. (K)	2162	2226	2283
Maximum Field Temp (K)	1557	1591	1650
Maximum NOx conc. (PPM)	214	396	587
Exit NOx conc. (PPM)	138	285	482

Table 2: Summary of NOx Parametric Study 1

Another point worth noting concerns the flame aerodynamics in the furnace as depicted in Figures 13 to 15. As the air temperature is increased, the extent of both internal and external recirculation zones is enhanced significantly. Internal recirculation is generated by the swirler and is responsible for the stabilisation of the flame, thus suggesting a better flame stability as air temperature is increased.

Although these are well known practical facts, the simulations were able to reflect their implications on the flame structure.

4.1.2 Influence of furnace geometry on flame structure

The same burner is now simulated in a rectangular furnace 9 ft. wide, 14 ft. wide and 26 ft. long, which is considerably larger than in the previous simulation.

The streamlines in Figures 16 and 17 show an increase in the radial expansion of the external recirculation region in the vertical direction of the furnace and causes significant departure from symmetry. However, the ignition zone structure (internal recirculation) remains almost unchanged. Temperature profiles across the furnace setion show clearly the flame asymmetry generated by the furnace confinement. (Figure 18)

The NOx contour plot given in Figure 19 shows a different structure from previous predictions for the cylindrical furnace. The maximum concentration is now well away from the burner and the mean NOx emissions at the exit, as a result, are reduced by 27%.

These tendencies were known from experimental evidence however no satisfatory explanations can be given at this stage.

4.1.3 Influence of dual-burner firing on flame aerodynamics

This study has great practical implications since most of the large industrial steam generators use an array of burners, from 2 to 10 in the mid-size, up to 32 burners for the 750 MW utility boilers.

The velocity vectors in a cross-section located 4.76 m downstream are shown in Figure 20 and provide information on the interaction between adjacent burners.

The interaction affects the fuel consumption rate where the flames overlap as can be seen from Figure 21.

Unfortunately, NOx emission runs were not completed when this paper was written. However, it is obvious from the early results that the number of burners and burner arrangement in a given furnace will strongly influence NOx emission predictions.

4.2 Peripheral Fuel Injection Mode

In this case, the burner uses six axial spuds located on a radius immediately outside the swirler to inject the fuel. The spud geometry allows the gas to be injected with a variable tangential and radial velocity component, while the axial component is maintained constant. This design along with its multitude of variations is used in practice when shorter flames are desired to satisfy furnace depth limitations. Amongst the design variations, the annular injector or "gas ring" falls into this category.

As shown in Figure 22, the flame structure is significantly altered. The fuel concentration is much lower (near stoichiometry) in the stabilization zone than with the central injector mode. This will have major consequences on temperature and NOx concentrations. In addition, the fuel depletion rate is faster and results, as expected, in a shorter flame.

Temperature contours (Figure 23) show higher maxima in the stabilization zone and in the flame periphery. This actually explains the fact that this flame is usually more stable at high firing rates. It also indicates that thermal NOx emissions will be higher than with the central fuel injection mode. This was also experienced during field tests where an approximate 30% NOx emission increase was observed.

Although this design is easier to deal with for flame stability and length considerations, low NOx burner designers would be advised to treat it with caution.

V- DISCUSSION OF RESULTS

The partial results of this study show that the design of low NOx combustion equipment must take into consideration the combustion chamber geometry. It also demonstrates that NOx emission is intimately related to the flame structure.

The flame structure was seen to be governed mainly by the fuel injection mode and the aerodynamics prevailing inside the furnace.

The partial results obtained so far in this investigation suggest that the central fuel injection where a fuel rich core is created in the flame front is well adapted for low NOx operation. However, some stability problems may arise when fuel concentration in the internal recirculation becomes too high to sustain combustion. This phenomenon was experienced in the field and was solved by the introduction of a small quantity of air flowing through the co-axial inner conduit of the gas injector. This model derived design alteration has proved to be very effective in the field in improving the flame stability, while leaving NOx emissions unchanged.

VI- CONCLUSIONS

The present mathematical model is a powerful design tool to assist the burner designer in predicting NOx emission trends. It allows comparative evaluations to be successfully performed on a series of burner design variations.

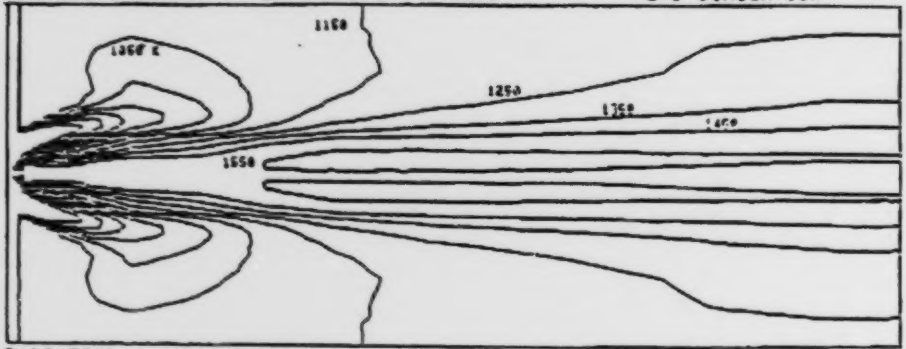
However, more experimental work in conjunction with the present model is required in order to predict absolute values of NOx emissions on specific applications. Particularly, a fuel-NOx model and an oil droplet combustion model must be incorporated to widen the application range of the present work.

Nonetheless, the preliminary results partially presented here were sufficient, with the help of some empirical burner design guide rules, to accomplish major improvements in the NOx emission performances of recent burner installations. For instance, the modelling results have suggested the effectiveness of a selective flue gas recirculation concept through the burner. Such a flue gas recirculation system was designed accordingly and implemented in a refinery early in July 1987. Impressive NOx reduction levels (80%) were achieved with a relatively small amount of the gas being recirculated (less than 16% of total). This was already a significant contribution to better burner design.

Empirical burner design methods have long reached their limits. But in the light of mathematical modelling work, new audacious burner designs are merging to meet todays and future environmental constraints.

PARAMETRIC STUDY 1

DYNASWIRL BURNER : TEMPERATURE CONTOURS 2-D SIMULATION



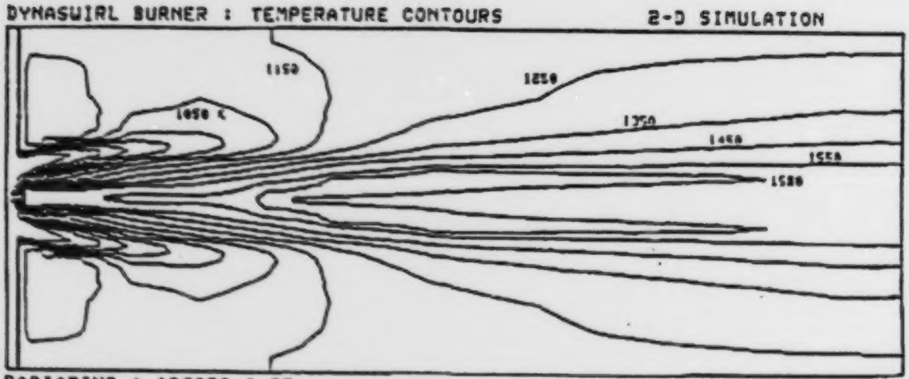
RADIATING : ABSORC=0.25
: SCATCO=0.005
KINETICALLY CONTROLLED COMBUSTION
CENTRAL FUEL INJECTION

BURNER OUTPUT : 100 MM BTU/HR
COMB AIR TEMP : 70 F
BURNER RDL : 10 in w.g.
FURNACE SIZE : 9ft diam. X 20ft long
FURNACE WALL : 500 F
EXCESS AIR : 8.14%

Figure 7

PARAMETRIC STUDY 1

DYNASWIRL BURNER : TEMPERATURE CONTOURS 2-D SIMULATION



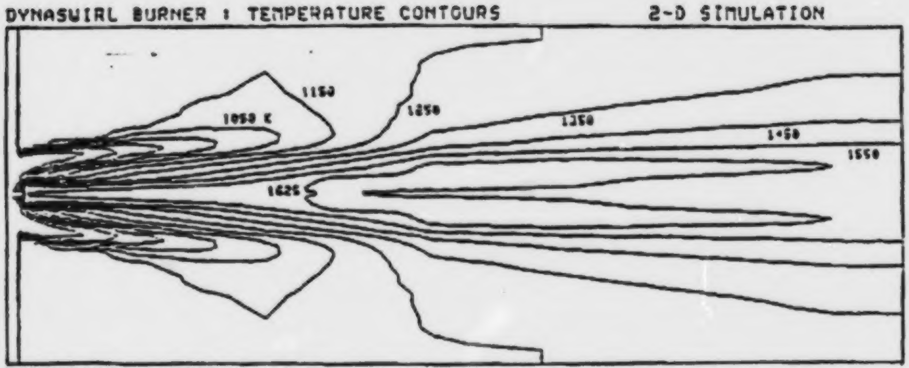
RADIATING : ABSORC=0.25
: SCATCO=0.005
KINETICALLY CONTROLLED COMBUSTION
CENTRAL FUEL INJECTION

BURNER OUTPUT : 100 MM BTU/HR
COMB AIR TEMP : 300 F
BURNER RDL : 10 in w.g.
FURNACE SIZE : 9ft diam. X 20ft long
FURNACE WALL : 500 F
EXCESS AIR : 8.14%

Figure 8

PARAMETRIC STUDY 1

DYNASWIRL BURNER : TEMPERATURE CONTOURS 2-D SIMULATION



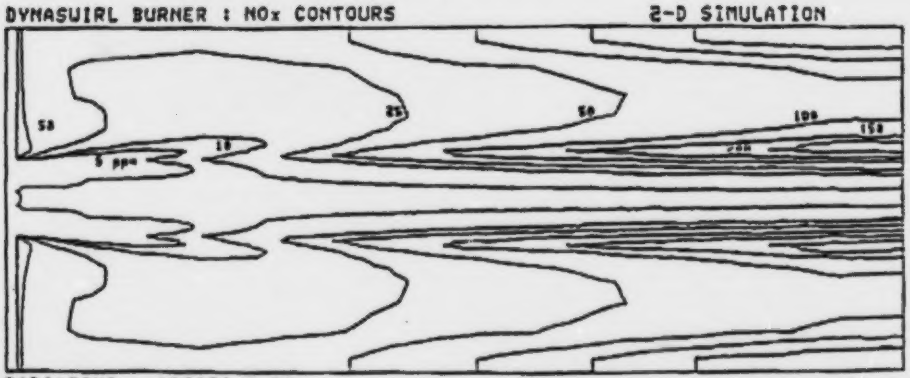
RADIATING : ABSORC=0.25
: SCATCO=0.005
KINETICALLY CONTROLLED COMBUSTION
CENTRAL FUEL INJECTION

BURNER OUTPUT : 100 MM BTU/HR
COMB AIR TEMP : 500 F
BURNER RDL : 10 in w.g.
FURNACE SIZE : 9ft diam. X 20ft long
FURNACE WALL : 500 F
EXCESS AIR : 8.14%

Figure 9

PARAMETRIC STUDY 1

DYNASWIRL BURNER : NOx CONTOURS 2-D SIMULATION



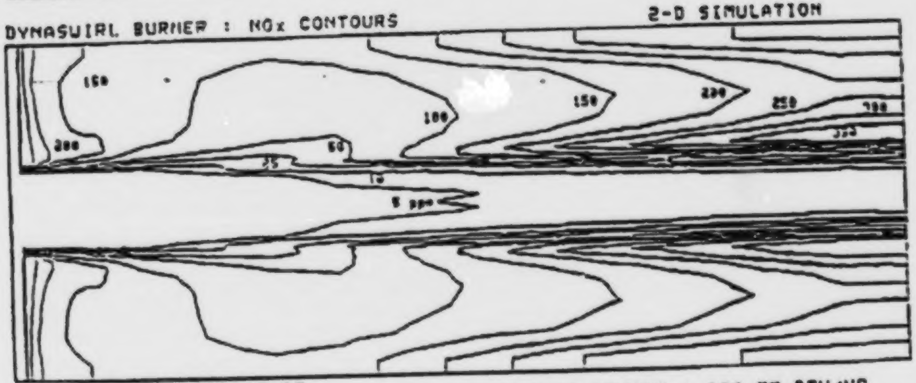
RADIATING : ABSORC=0.25
: SCATCO=0.005
KINETICALLY CONTROLLED COMBUSTION
CENTRAL FUEL INJECTION

BURNER OUTPUT : 100 MM BTU/HR
COMB AIR TEMP : 70 F
BURNER RDL : 10 in w.g.
FURNACE SIZE : 9ft diam. X 20ft long
FURNACE WALL : 500 F
EXCESS AIR : 8.14%

Figure 10

PARAMETRIC STUDY I

DYNASWIRL BURNER : NOx CONTOURS 2-D SIMULATION



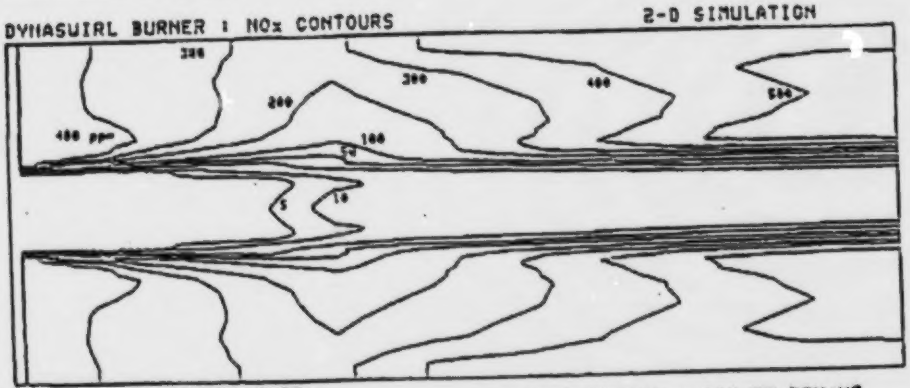
RADIATING : ABSORC=0.25
: SCATCO=0.005
KINETICALLY CONTROLLED COMBUSTION
CENTRAL FUEL INJECTION

BURNER OUTPUT : 100 MM BTU/HR
COMB AIR TEMP : 300 F
BURNER RDL : 10 in w.g.
FURNACE SIZE : 9ft diam. X 20ft long
FURNACE WALL : 500 F
EXCESS AIR : 8.14%

Figure 11

PARAMETRIC STUDY I

DYNASWIRL BURNER : NOx CONTOURS 2-D SIMULATION



RADIATING : ABSORC=0.25
: SCATCO=0.005
KINETICALLY CONTROLLED COMBUSTION
CENTRAL FUEL INJECTION

BURNER OUTPUT : 100 MM BTU/HR
COMB AIR TEMP : 500 F
BURNER RDL : 10 in w.g.
FURNACE SIZE : 9ft diam. X 20ft long
FURNACE WALL : 500 F
EXCESS AIR : 8.14%

Figure 12

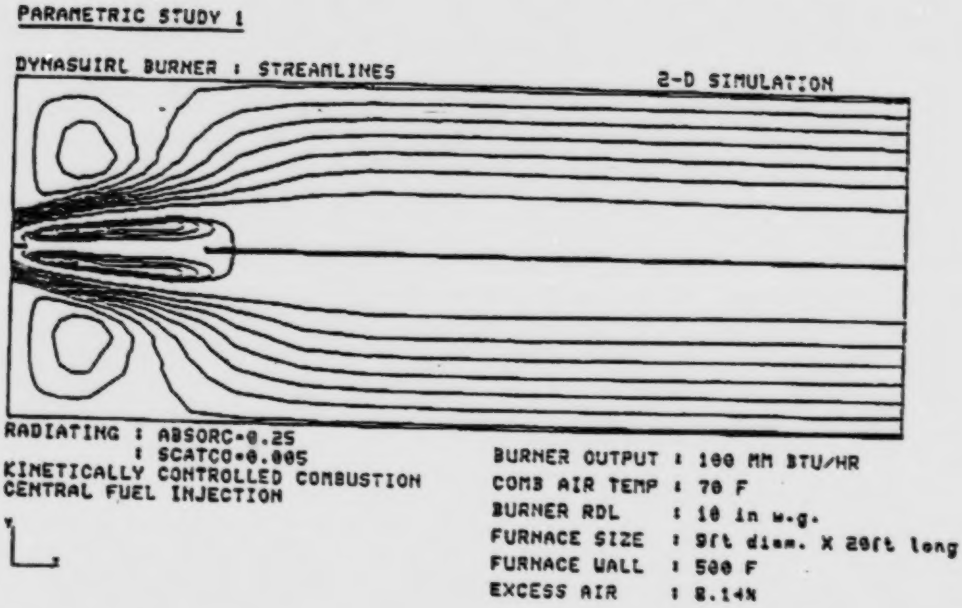

PARAMETRIC STUDY 1
DYNASWIRL BURNER : STREAMLINES
2-D SIMULATION
RADIATING : ABSORC=0.25
: SCATCO=0.005
KINETICALLY CONTROLLED COMBUSTION
CENTRAL FUEL INJECTION
BURNER OUTPUT : 100 MM BTU/HR
COMB AIR TEMP : 70 F
BURNER RDL : 10 in w.g.
FURNACE SIZE : 9ft diam. X 20ft long
FURNACE WALL : 500 F
EXCESS AIR : 8.14%

Figure 13

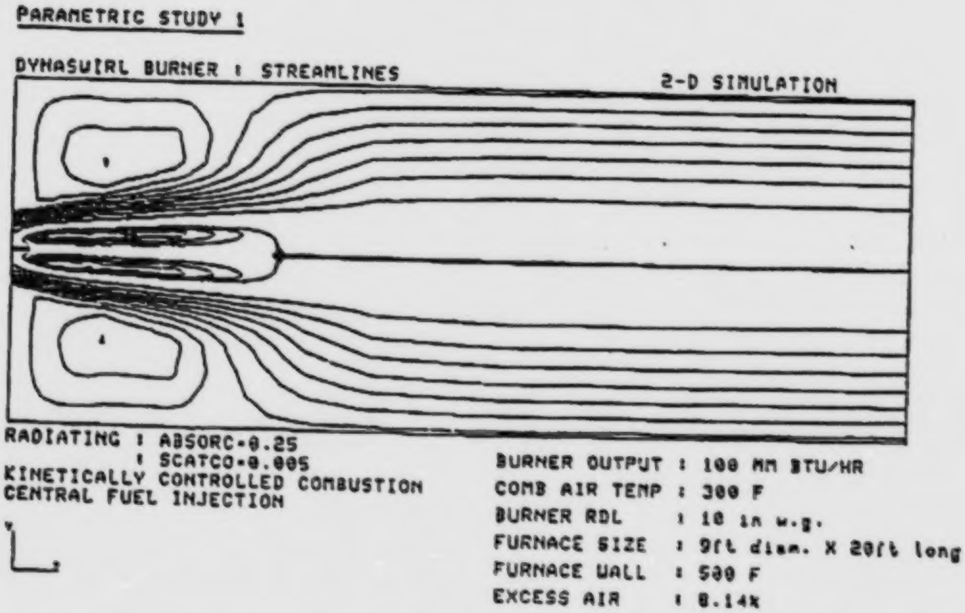

PARAMETRIC STUDY 1
DYNASWIRL BURNER : STREAMLINES
2-D SIMULATION
RADIATING : ABSORC=0.25
: SCATCO=0.005
KINETICALLY CONTROLLED COMBUSTION
CENTRAL FUEL INJECTION
BURNER OUTPUT : 100 MM BTU/HR
COMB AIR TEMP : 300 F
BURNER RDL : 10 in w.g.
FURNACE SIZE : 9ft diam. X 20ft long
FURNACE WALL : 500 F
EXCESS AIR : 8.14%

Figure 14

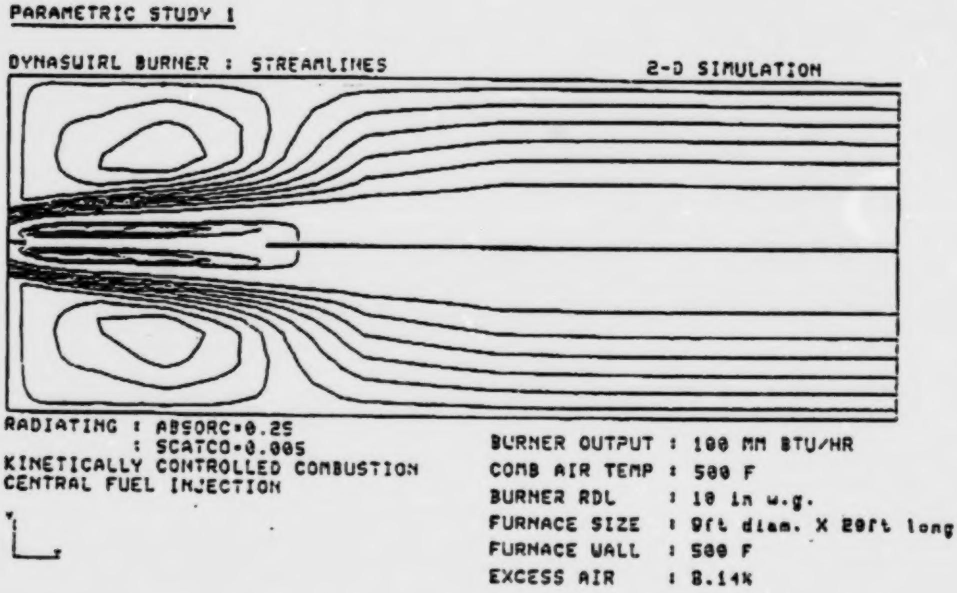


Figure 15

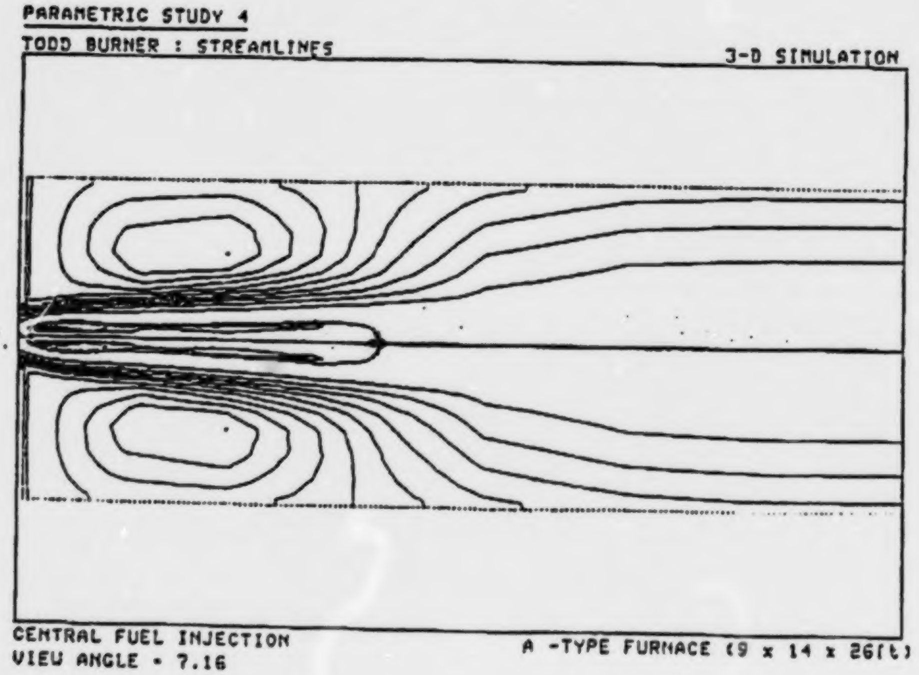


Figure 16

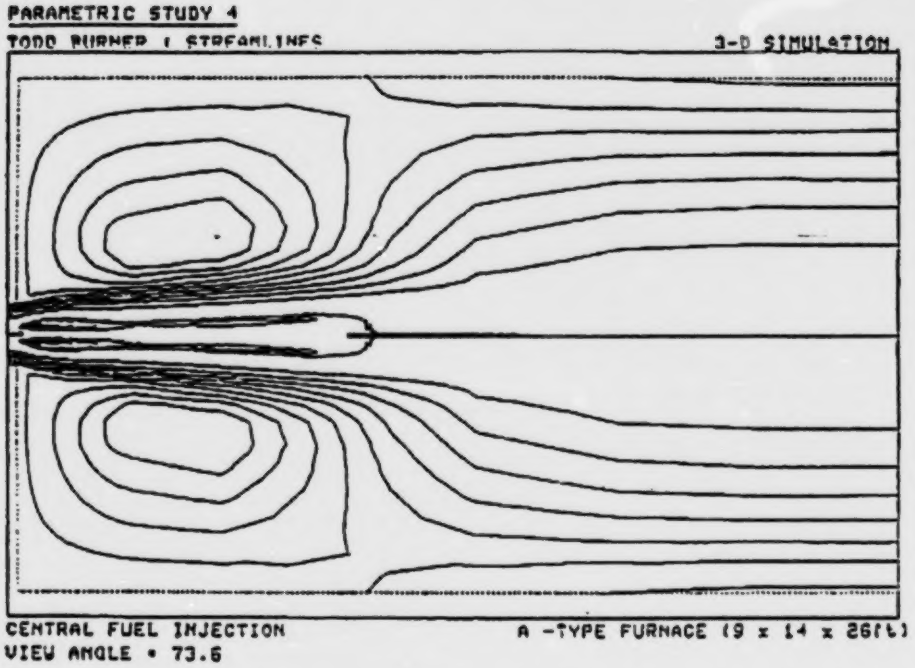


Figure 17

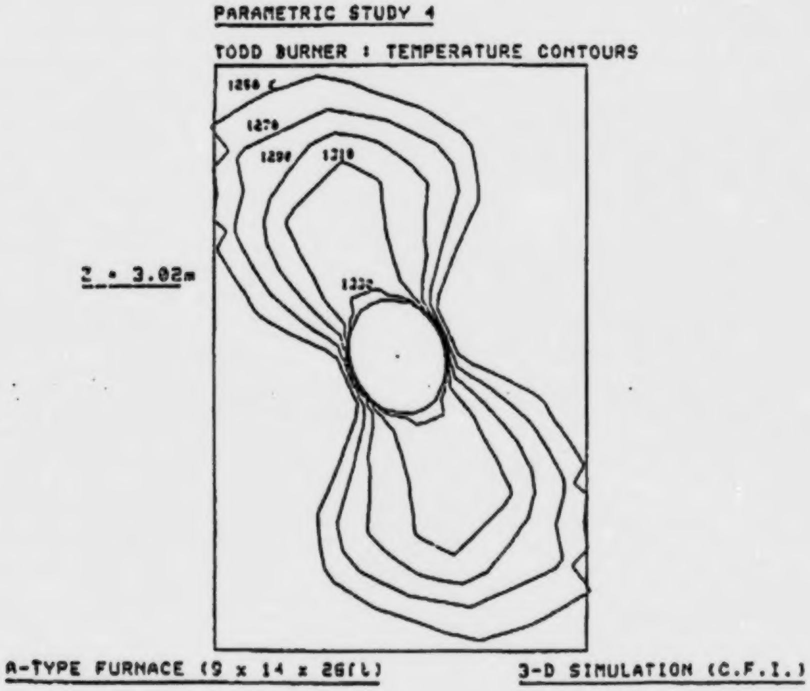


Figure 18

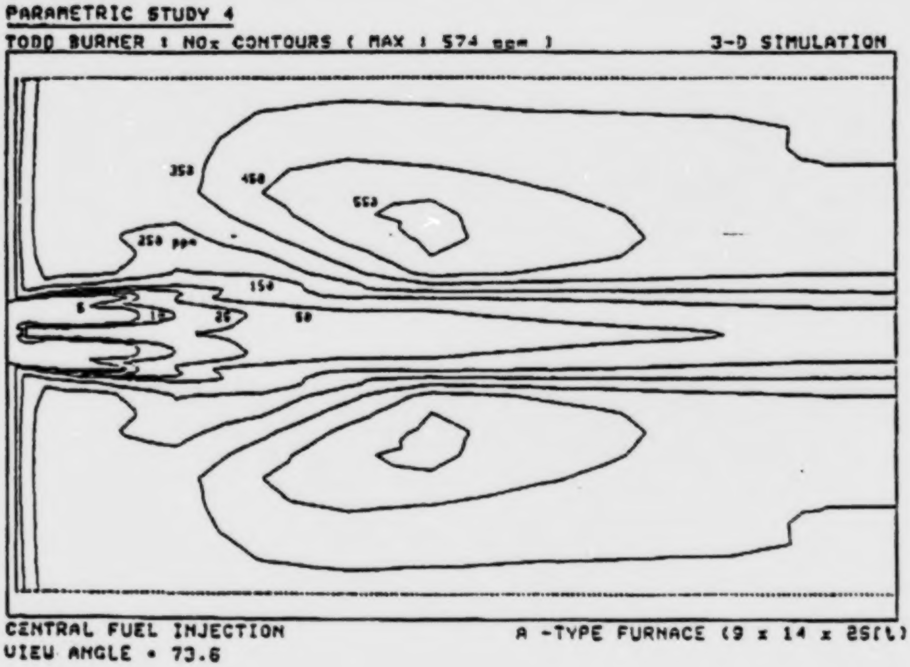


Figure 19

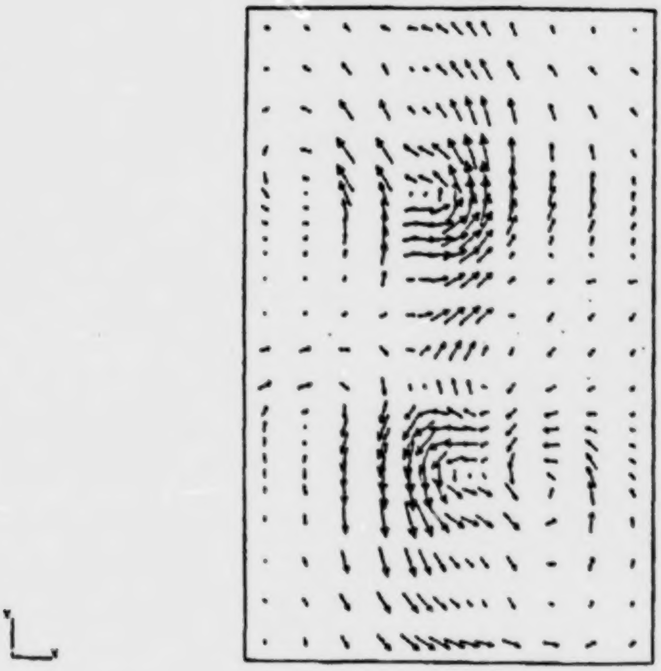


Figure 20

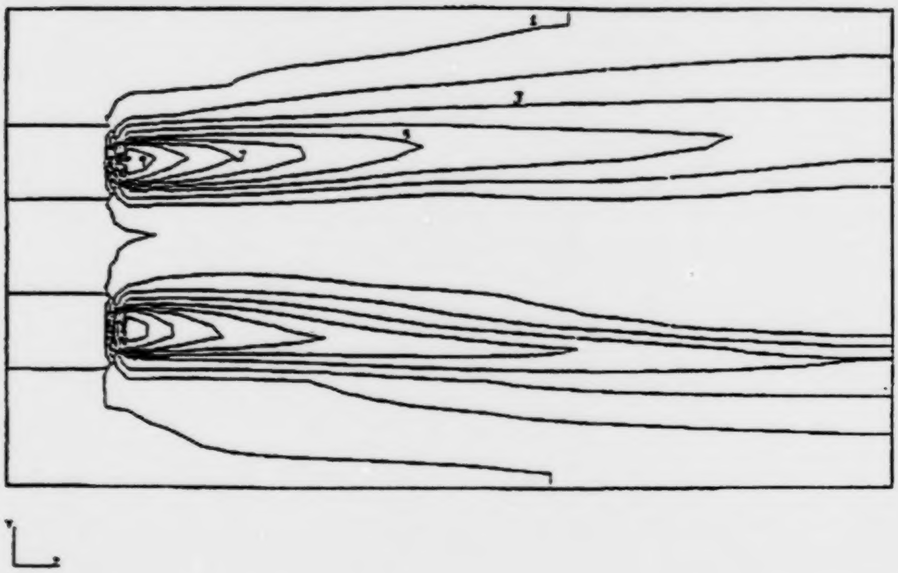
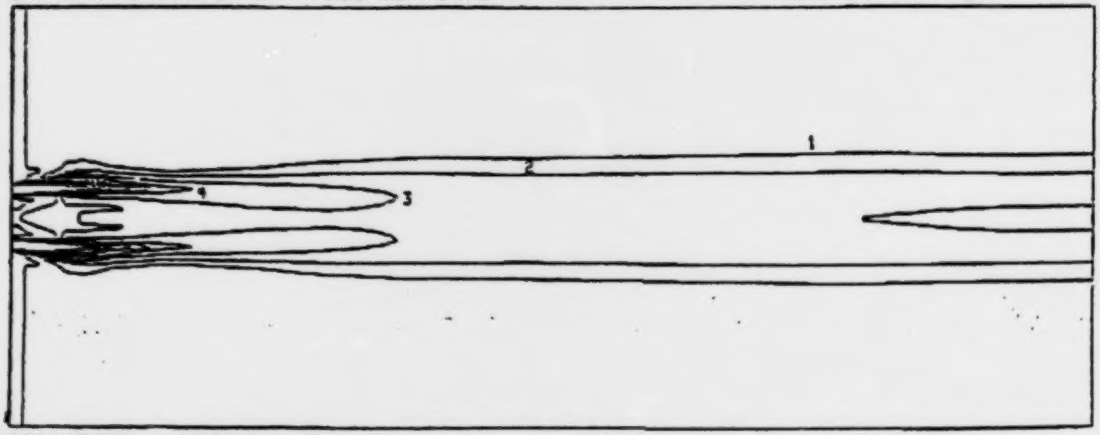


Figure 21

DYNASWIRL BURNER , FUEL CONTOURS 3-D SIMULATION

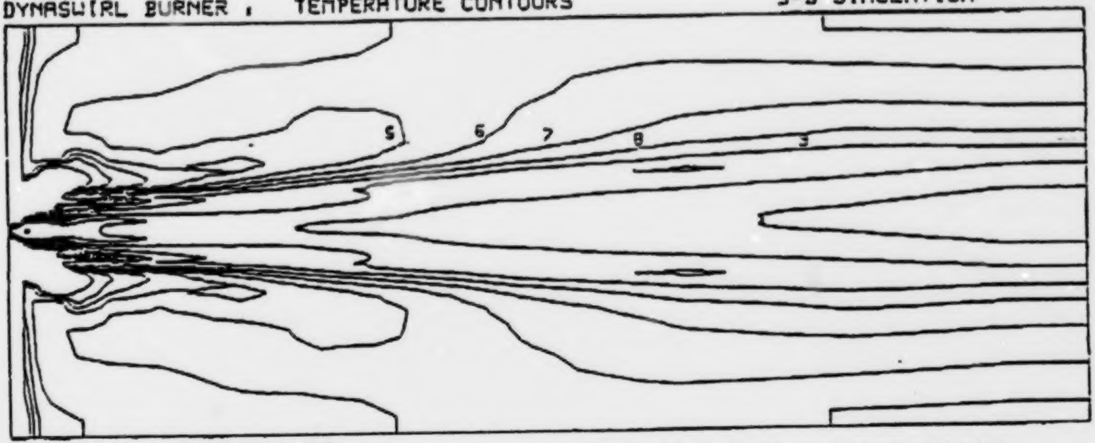


RADIATING , ABSORC=0.25
, SCATCO=0.005
KINETICALLY CONTROLLED COMBUSTION
CONTOUR RANGE , 0.005,(10),0.205

BURNER OUTPUT , 100 MM BTU/HR
COMB AIR TEMP , 500 F
BURNER RDL , 10 in w.g.
FURNACE SIZE , 9ft diam. X 20ft long
FURNACE WALL , 500 F
EXCESS AIR , 5 %

Figure 22

DYNASWIRL BURNER : TEMPERATURE CONTOURS 3-D SIMULATION



RADIATING : ABSORC=0.25
: SCATCO=0.005
KINETICALLY CONTROLLED COMBUSTION
CONTOUR RANGE : 980.(10).1980

BURNER OUTPUT : 100 MM BTU/HR
COMB AIR TEMP : 500 F
BURNER RDL : 10 in w.g.
FURNACE SIZE : 9ft diam. X 20ft long
FURNACE WALL : 500 F
EXCESS AIR : 5 %

Figure 23

REFERENCES

[1] R.F. Sawyer (1972)
'Atmosphere Pollution by Aircraft Engines and Fuels'
AGARD Advisory report 40.

[2] Energy World, June 1987: No. 148.

[3] Process Engineering, March 1987p. 41.

[4] A.H. Lefebvre (1983)
'Gas Turbine Combustions' McGraw Hill, p. 481.

[5] Shefer R.W. and Sawyer R.F. (1976)
'Lean Premixed Recirculating Flow Combustion for Control of Oxides of Nitrogen'

[6] Quan V, Kliegel J.R. and Peters R.L. (1975)
'Predicted Effects of Fluid Dynamic Parameters on Nitric Oxide Formation in Turbulent Jet Diffusion Flames'
Combustion and Flame 25, p. 67.

[7] Mizutani Y and Katsuki M. (1976)
'Emissions from Gas Turbine Combustors' Bulletin of JSME 19, p. 1360

[8] Khalil E.E. (1981)
'Numerical Computation of Combustion Generated Pollutants'
Proc. Italian Flames Days (La Rivista du Combustibili).

[9] Hutchinson P. Khalil E.E. and Whitelaw J.H. (1977)
'Measurement and Calculation of Furnace Flow Properties'
AIAA Journal of Energy 1, p. 212.

[10] Hung W. (1975)
'An Experimentally Verified Nox Emission Model for Gas Turbine Combustors, ASME 75-GT-71

[11] Zeldovich Y.B., Sadovinkov P.Y. and Frank-Kamenctskii D.A. (1974)
'Oxidation of Nitrogen in Combustion'
Academy of Sciences of USSR, Moscow

[12] Markatos N.C. Malin M.R. and Cox G. (1982)
'Mathematical Modelling of Buoyancy-IOnduced Smoke Flow in Enclosures'
Int.J. Heat and Mass Transfer 25, pp. 63-75.

[13] Rodi W. (1978)
'Turbulence Models and their Applications in Hydraulics - a State of the Art Review'
SFB 80/T/127, Univ. of Karlsruhe

[14] Spalding, D.B. (1977)
'GENMIX: A General Computer Program for Two-Dimensional Parabolic Phenomena'
Pergamon Press.

[15] Markatos N.C. and Pericleous K.A. (1983)
'An Investigation of Three-Dimensional Fires in Enclosures'
ASME - MTD 25, pp. 115-124.

[16] Magnussen B.F. and Hjertager B.H. (1976)
'On Mathematical Modelling of Turbulent Combustion with Special Emphasis on Soot Formation and Combustion'
Proc. 16th Int. Symposium on Combustion. The Comb. Inst., pp. 719-729.

[17] Hamaker H.C. (1947)
'Radiation and heat Conduction in Light Scattering Material',
Philips Res. Rep. 2, pp. 103-111

[18] Gordon S. and McBride B. (1971)
'CREK: A Computer Program for Calculations of Complex Chemical Equilibrium Compositions' NASA SP-273.

[19] Siegmunc C.W. and Turner D.W. (1974)
'NOx Emissions from Industrial Boilers: Potential Control Methods',
Journal of Engineering for Power, Jan., p. 1

[20] Lisauskas R.A. and Snodgrass R.J. et al (1985)
'Experimental Investigation of Retrofit Low NOx Combustion Systems' Joint Symposium on Stationery Combustion NOx Control, May 6-9, Boston, 1985.

[21] DeSoete G. (1977)
'Etude Parametrique des Effets de la Stratification de la Flamme Sur les Emissions d'Oxydes d'Azote' Revue de l'Institut Français de Pétrole 32(3), p. 427.

[22] Gupta A.K. Liley D.G. and Syred N. (1984)
'Swirl Flows' Abacus Press.

SECTION

VIII

COMPUTER SIMULATIONS FOR INDUSTRIAL PROCESSES

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Kenji Ushimaru
Energy International Inc.

9/5/666

ABOUT THE SPEAKER:

Kenji Ushimaru holds MS and BS degrees in Aeronautics and Astronautics from the University of Washington.

From 1978 to 1980, Kenji was a Research Assistant at the University of Washington and then joined Flow Industries Inc. as a Research Engineer.

He joined Energy International Inc. in 1983 to become Program Manager, Industrial Processes, the position he now holds.

Kenji's special areas of interest include fluid mechanics, multi-phase flows, and numerical simulation of energy and environmental systems. He is the co-holder of a U.S. Patent "Cogeneration Microwave Food Dryer".

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COMPUTER SIMULATIONS
FOR INDUSTRIAL PROCESSES

Presented at:

Industrial Gas Technology
Conference and Exhibition
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By:

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ABSTRACT

A numerical methodology has been developed for the design and analysis of gas-fired combustion systems. The simulation technique is based on a three-dimensional fluid dynamics analysis program which is capable of numerically simulating fuel-air mixing, reaction kinetics, heat transfer, and time-dependent flow situations over complex geometries. The present simulation technique can treat a wide variety of boundary conditions, is numerically stable at all Reynolds numbers, and can be applied to a broad class of burner and furnace geometries and associated flows. Examples are presented which demonstrate its use as an effective engineering tool for the analysis of key performance parameters in industrial applications such as heat transfer effectiveness, emission control, combustion efficiency, and flame stability.

1. Introduction

A significant fraction of on-line industrial burners use natural gas as a preferred fuel. Boilers alone, for example, account for more than 40 percent of the total gas consumed by the U.S. industrial sector. It follows, then, that a key element in the commercial success of industries using gas-fired thermal processing systems is a properly designed burner. Moreover, stringent demands are placed on modern burner designs since industrial users also require other significant advantages, such as higher productivity, improved energy efficiency, and reduced pollutant emissions.

Unfortunately, the basic physical processes of mixing and chemical reaction are highly coupled for most fuels. Moreover, the physical interaction of injectors, walls, heat exchangers, reactant inlet conditions, and fuel chemical properties strongly affect burner performance. Without reliable engineering tools, combustion and heat transfer equipment has, for the most part, been designed by costly trial-and-error methods. Although the industry has long recognized the need to develop numerical descriptions of combustion processes, it is only in recent times that advances in computer hardware and computational methods have permitted the problem to be addresses in earnest.

The objective of this presentation is to describe the development of a numerical simulation code which is suitable for the analysis of many kinds of thermal processing systems. The code, which can accurately model time-dependent turbulent flow fields with recirculation, is an advanced fluid dynamics and heat transfer tool which can be utilized to analyze the design of direct and indirect gas-fired burners, radiant tube burners, coal-gas or oil-gas co-firing burners, and a variety of low NO_x burner techniques.

A synopsis of the numerical model and examples of its application to industrial thermal processes follow directly.

2. Simulation Model

The technique is based on physical principals for analysis of incompressible, three-dimensional, steady and transient flows with turbulence. Presently, a family of codes for a variety of problems has been derived, which is named the "PS-family" to indicate "process simulators" of codes. Presently, the three basic PS codes described in Table 1 are available.

Fluid dynamics of flows with irregular boundaries may be described since the technique employs a general three-dimensional (nonorthogonal) coordinate system with a boundary fitted mesh. In addition to solving the conservation equations of mass, momentum, and energy, the code uses a two-equation turbulence model to describe the transport of turbulent kinetic energy. Many different boundary conditions can be accommodated, and the solution technique is numerically stable for all Reynolds numbers. The theoretical basis and the details of the PS family of codes are well documented in the literature (Ref. 1-7), and the basic numerical approach is discussed below.

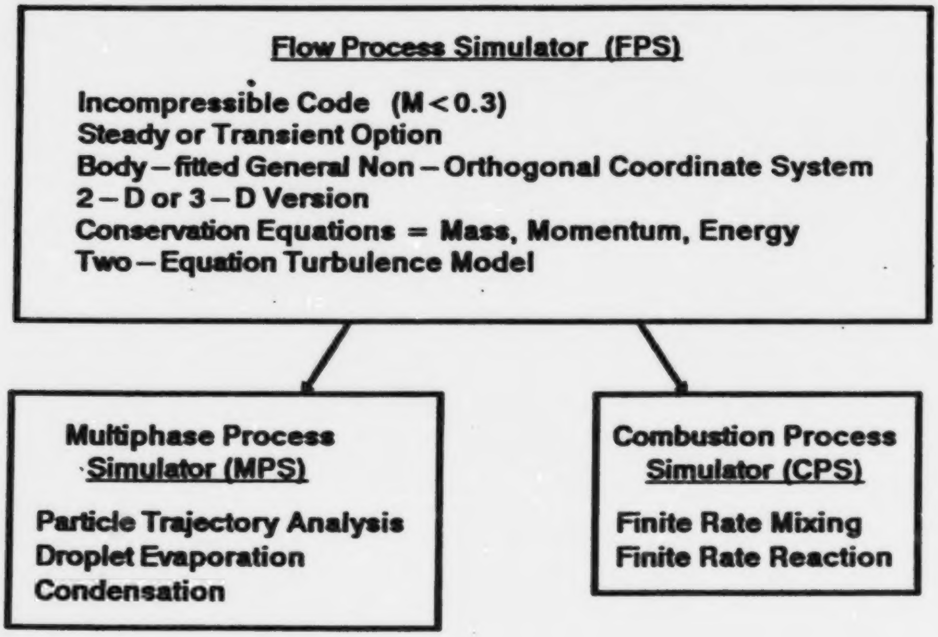
3. Finite Volume Approach

The numerical model uses finite volume methods to descritize the transport equations. A typical computational control volume, with its associated coordinate axes, is sketched in Figure 1. Capital letters designate centers of adjacent control volumes, and lower case letter indicate the midpoints of each control surface. A grid generation scheme is used to divide the solution domain into a finite number of these volumes with the values of scalar quantities (e.g., pressure and temperature) associated with the center of each volume. In the previous developments (Reference 1-7), the component velocities had to be defined at the midpoint of the control surfaces (in a staggered manner). The resultant mesh, called the staggered mesh system, had critical analysis parameters, such as x-velocity, y-velocity, and z-velocity, at points 1/2 cell away from the central node where temperature and pressure were defined. With many of presently available codes, the staggered mesh system is necessary for viscous, incompressible flow solvers in order to improve the numerical stability of the pressure calculation. While the staggered mesh system, with its numerical stability, has been a preferred approach, it does complicate the coding and interpretation of the predicted results.

In an effort to simplify the coding and extraneous steps necessary for a user to interpret the computed results, the PS-family of codes has been developed on the concept that all variables are computed at common locations centered in the mesh volume. The non-staggered (central) location is also complemented by an improved pressure solver to alleviate the inherent stability problem. Thus, the PS-family of codes established a means by which the code is much simpler to understand and its predicted results much easier to interpret for the user, while maintaining good numerical stability.

Table 1. Code/Application Matching Technique

FPS – Flow Process Simulator
MPS – Multiphase Process Simulator
CPS – Combustion Process Simulator



Flow Process Simulator (FPS)

- Incompressible Code (M < 0.3)
- Steady or Transient Option
- Body – fitted General Non – Orthogonal Coordinate System
- 2 – D or 3 – D Version
- Conservation Equations = Mass, Momentum, Energy
- Two – Equation Turbulence Model

Multiphase Process Simulator (MPS)

- Particle Trajectory Analysis
- Droplet Evaporation
- Condensation

Combustion Process Simulator (CPS)

- Finite Rate Mixing
- Finite Rate Reaction

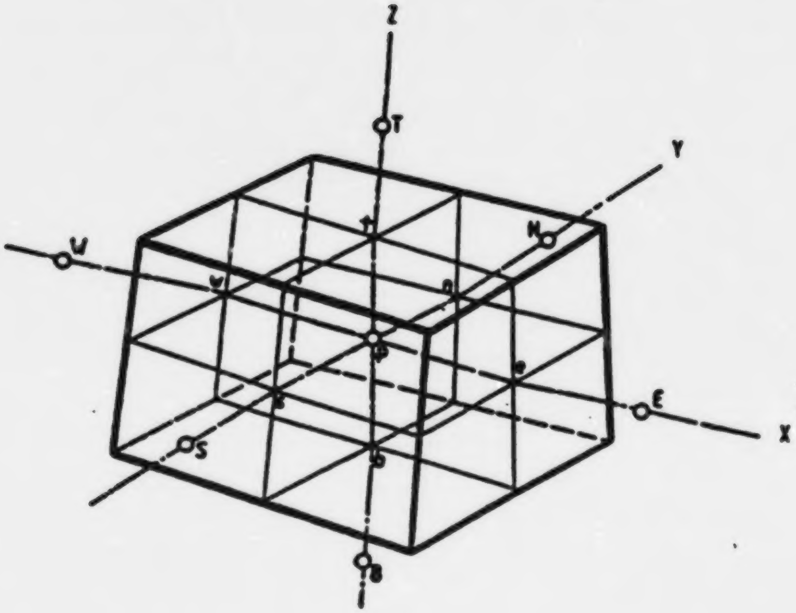


Figure 1. Computational Control Volume Showing Coordinate Directions (X,Y,Z), Points of Definition for Scalers (P), Velocity Components (n, s, e, w, t, b), and Adjacent Control Volume Centers (N, S, E, W, T, B).

Another noteworthy benefit arising from the grid system employed by the PS-family of codes is ease of program interface with complex geometries. The PS-family of codes is based on the Cartesian coordinate system and all central cell nodes can be denoted and identified by the familiar (x,y,z) notation with corresponding velocity components (u,v,w). This system conveniently avoids the need for algebraic velocity mapping or transformation, and the resultant vectors can be plotted for immediate flow visualization by the user. Furthermore, the use of the Cartesian-based vectors eliminates the curvilinear momentum source terms (e.g., Coriolis and centrifugal force terms) existing in general curvilinear coordinates. Hence, the convergence problems usually associated with the analysis of highly swirling flows are eliminated.

4. Solution Procedure

To linearize the differential equations, a hybrid central/upwind differencing scheme is applied to the convection and diffusion terms. The finite difference equations are solved by integrating each variable over the control volume in which the variable is defined. Fluid properties outside the associated control volumes are determined by linear interpolation. Pressure and velocity are coupled through the continuity and momentum equations. These equations are solved using the fully implicit Pressure Implicit Split Operator method (PISO). This algorithm solves the finite-difference equations by a cyclic series of predictor and corrector operations, wherein the velocity solutions are corrected to satisfy mass conservation through adjustments to the pressure field. At each predictor and corrector step, the dependent variables, including pressure increments, are determined using an efficient line-by-line tri-diagonal matrix solver (TDMA). The grid is swept in alternate directions to obtain the solution over the entire physical domain.

The major advantage of PISO is an improved rate of convergence over the more commonly used SIMPLE algorithm (Ref.3). Numerical evaluations indicate that PISO is faster than SIMPLE by 2-6 times for steady-state problems, and as much as 10 times for transient flow problems (Ref.1).

5. Convergence Criteria

Two criteria are used to monitor the solution convergence. The first is a fractional change criterion in which the fractional change between successive iterations is examined over the entire set of computational nodes. The maximum value of the fractional changes is required to be less than a prescribed value. This criterion is a necessary but not a sufficient condition for convergence, since it may be satisfied when changes in flow parameters are small. Fore this reason, a residual source criterion is also applied. The summation of the normalized residuals taken over the entire computational domain is also required to be less than a preset value.

6. Application Know-How

Today a user who may be contemplating a purchase or lease of a flow simulation code is faced with a difficult choice -- what flow simulator to try out and on what computer workstation. In today's market for both numerical codes and computer workstations, a user can find a dozen candidates for flow simulation codes for any given type of problem and perhaps five workstation candidates.

Undoubtedly, it would require a significant effort to select the "right code" and the "right workstation" for the type of problems to be solved. The situation is further complicated by the wide range of prices for both software packages and hardware products (and their options) which are offered. In today's simulation code market, therefore, good know-how in analyzing code-application matching is useful.

From the user's standpoint, a checklist of analysis requirements is recommended. The checklist, such as the one shown in Table 2, is a way to describe what type of flow situations are most often encountered by the user. The user should first develop a set of detailed specifications for his problem needs before comparing various codes available on the market. The checklist starts from obvious choices, such as steady or transient, problem dimensionality, compressibility, and temperature calculation. More subtle differences can be characterized by whether particles in gas or liquid droplets in gas can be simulated, or whether an algebraic non-Newtonian fluid model is optionally available (for simulation of soap flow or chemical agents used in oil fields, for example).

The next step is to test the code; this should be conducted in concert with a workstation vendor. Although today's state-of-the-art workstations are computationally efficient and are based on fast, cost-effective computer chips, such differences as the FORTRAN compiler, graphics capability, and internal memory (CPU) size can make a marked difference in computing speed of identical codes being run on different machines. The user can try out some select candidate codes on a limited trial-basis to determine ease of problem setup, computing efficiency, numerical stability, and interpretation of computed results as a benchmark study to reach the right conclusion.

7. Simulations of Industrial Processes

The PS codes have been used to simulate the fluid dynamics and heat transfer resulting from three example studies:

1. A utility boiler simulation over complex geometry.
2. Two coaxial jets expanding into a rectangular furnace.
3. Preliminary results of natural gas combustion in a furnace and associated NO_x calculation

Table 2. Checklist of Analysis Requirements

1. Analyze the problem

Temporal Dimension → Steady State or Transient
Physical Dimension → 2 – D, Axisymmetric, or 3 – D
Flow Reynolds Number → Viscous or Inviscid
Flow Mach Number → Compressible > M 0.3 > Incompressible
Turbulence Effect → Laminar or Turbulent Flow
Temperature Effect → Heat Transfer or Isothermal
Reaction Kinetics → Reacting Flow or Nonreacting Flow
Fluid Property → Newtonian or Non – Newtonian
Fluid Phase → Single Phase or Multiphase

2. Analyze the Code

All of the Above and:

- Ease of Problem Setup
- Memory Requirement
- Computing Efficiency
- Stability Range

These computational results are described briefly.

Utility Boiler Simulation

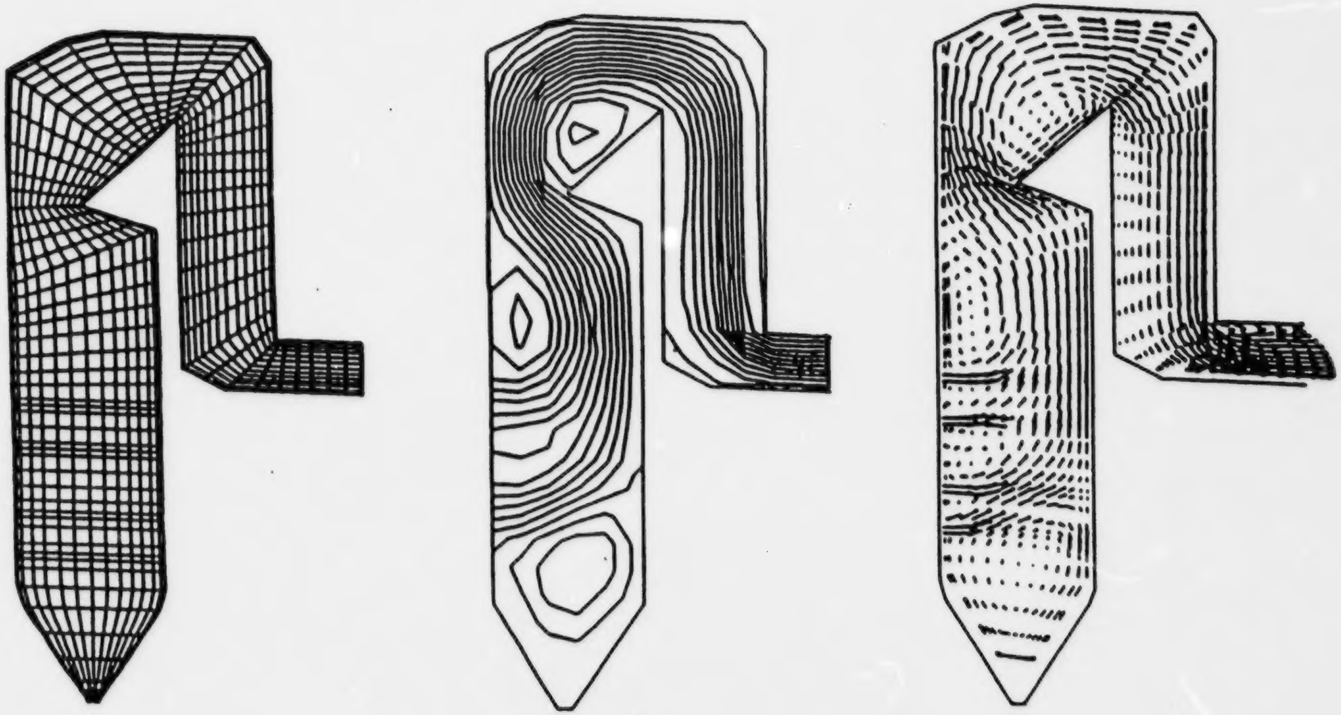
Complex flow fields such as the utility furnace flow field have many bends and sloping walls which are in most cases not co-linear with the coordinate directions (x,y,z). In general, the use of "dummy grids" and the crude approximation of sloping walls by stair-type grids are inevitable with the orthogonal-grid based codes. However, using the boundary-fitted computational grid system of the FPS code, the complex flow field can be analyzed economically and more accurately because no grids are wasted nor is the crude approximation necessary for the simulation task. The grid generator of the FPS code not only provides the boundary-fitted grids, but also improves the grid aspect ratio to minimize the numerical diffusion effect and improve stability for convergence. The two-dimensional flow analysis of an example utility boiler is shown in Figure 2. Figure 2-a illustrates the two-dimensional grids of the utility boiler system, which fits the boundary walls exactly. The boiler was 32 m high and 7 m wide at the main boiler section. The grid system consisted of 140x29 nodes (4060 computational cells). Flow condition at the burner injection ports was set at 22.8 m/s. Temperature was set at 600^{o}C and assumed constant (isothermal flow case) throughout the flow situation. Flow boundary conditions for the multiple burners (four on each side of the boiler in this case) satisfy the mass and momentum fluxes of the incoming fuel and air flows for individual burners.

Figure 2-b shows the overall streamlines as a result of the simulation. The simulation was conducted on an Apollo DN560 workstation with 3 MB of internal CPU and 75 MB of fast access disk. The calculation took 15.5 hours to converge after 712 iterations. CPU time per iteration and per node was estimated to be 19.2 msec. In Figure 2-b, two major recirculation zones are apparent. For applications such as chemical injection into the utility boiler for NO_x/SO_x control, the identification of possible recirculation zones and their approximate locations would be useful.

Figure 2-c illustrates the complex velocity field near the burner ports. Several secondary recirculation zones are observed, which indicate the varying degree of shear and mixing taking place as injected gas and air from the burner ports interact in the boiler. Comparison with field data taken in a utility boiler confirmed that the computed velocity field qualitatively agreed with the limited number of data points taken in the plant.

Coaxial Jet Mixing in a Furnace

The numerical model is especially suited to the prediction of wall-bounded flows with recirculation. It is used here to evaluate the importance of relative velocities between two co-flowing jets entering a mixing chamber (a furnace). This flow configuration is schematically shown in Figure 3. Combustion heated air and a cooler air shroud enter a closed chamber through 2-D slots, suddenly expand to the full chamber width, and exit through a sharp contraction. This flow geometry is also common to industrial combustion processes. Such processes include, but are not limited to the following:



2-a Two-Dimensional Grids

2-b Streamlines

2-c Velocity Vectors

Figure 2. Two-Dimensional Flow Analysis of a Utility Boiler

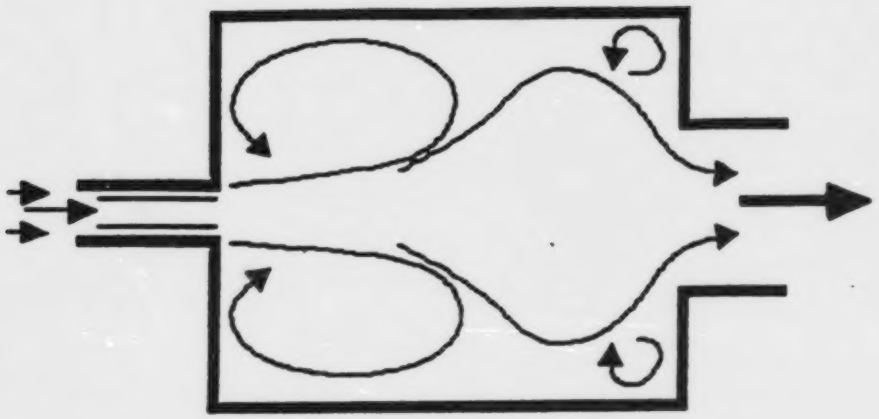


Figure 3. Two-dimensional Flow Geometry for Coaxial Jets Expanding Into a Cavity

1. Mixing and dilution of primary combustion products with secondary air to reduce pollutant emissions (NO_x and CO);

2. High-volume drying applications where large flow rates of warm air are required; and

3. High-intensity "can" or "dump" burners where the assumption of rapid chemical rates are appropriate.

The flow configuration shown in Figure 3 is central to many industrial processes which require large volumes of drying air or the reduction of primary combustion NO_x and CO emissions. If the chemical reactions are sufficiently fast, this flow model also provides first-order estimates of burner performance.

The primary stream enters the chamber through a 0.05 m slot with a velocity of 30 m/sec and temperature of 1700^{o}K. The secondary mass flow rate is held constant and its temperature set to 500^{o}K. The chamber walls are assumed adiabatic. The influence of the secondary stream inlet conditions is demonstrated by comparing the velocity, temperature, and heat flux distributions resulting from design choices of inlet velocity and injection height:

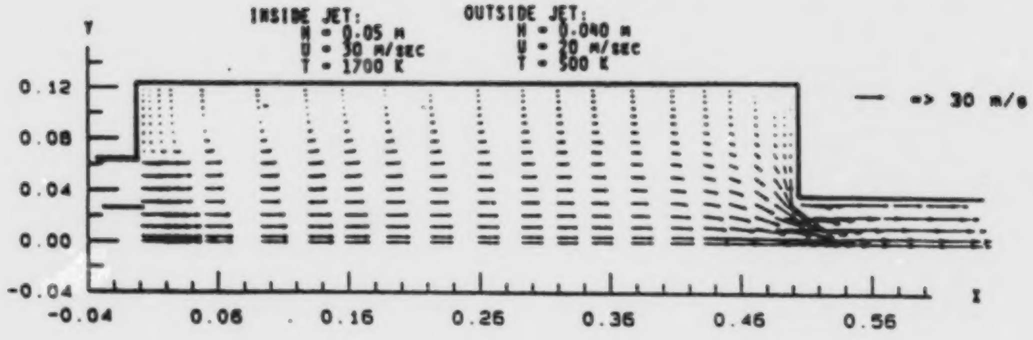
Case 1 Secondary air injection velocity of 20 m/sec and injection height of 0.04 m.

Case 2 Secondary air injection velocity of 40 m/sec and injection height of 0.02 m.

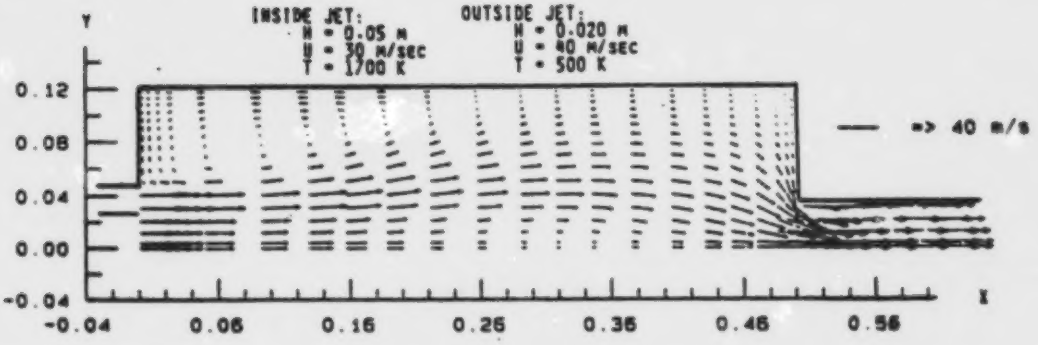
Note that the total system mass flow rate is the same for both cases.

Figure 4 presents a comparison of the velocity fields for Case 1 and 2. The major flow features are similar in character, with a large recirculation region near the wall just downstream of the inlet slots, and a small recirculation zone in the downstream corner at the contraction. There are, however, substantial differences between the two cases. The upstream recirculation zone of Case 2 (high velocity secondary air) extends approximately 40% further along the wall and has a markedly lower velocity region on the chamber centerline at approximately 0.30m. These differences result directly from the increased secondary jet momentum flux and larger initial flow expansion area.

The temperature contours and surface plots shown in Figure 5 and Figure 6 also illustrate important differences. When compared to Case 1, Case 2 exhibits greater transport of thermal energy away from the centerline and higher temperatures near the wall. Furthermore, Case 2 shows a more gradual variation in temperature across the chamber exhaust, indicating a higher mixing efficiency.



a) Slower Outside Jet



b) Faster Outside Jet

Figure 4. Velocity Field for 2-D Coaxial Jets Expanding into a Cavity

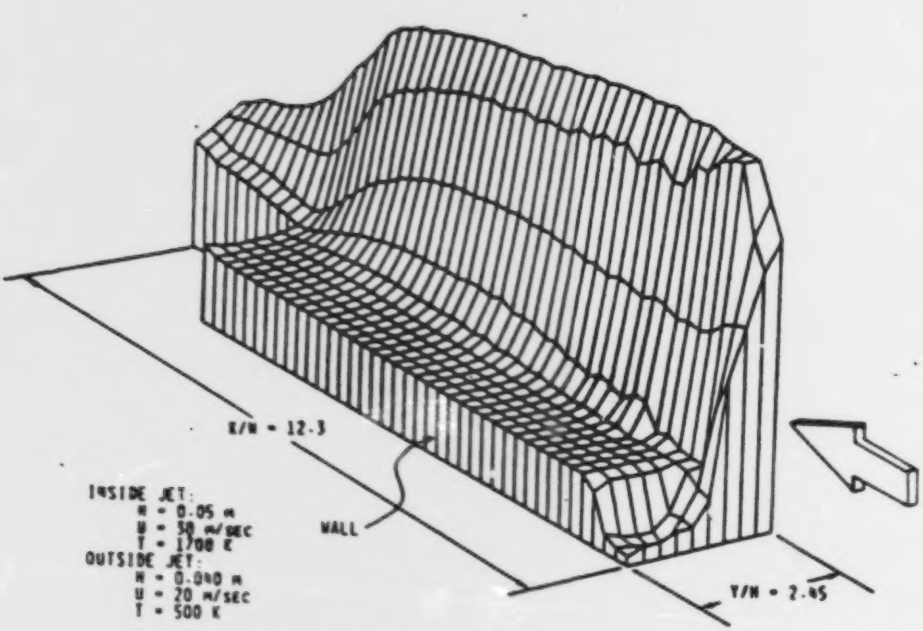


Figure 5. Contour Plot of Temperature Distribution for 2-D Coaxial Jets Expanding into a Cavity. Slower Outside Jet

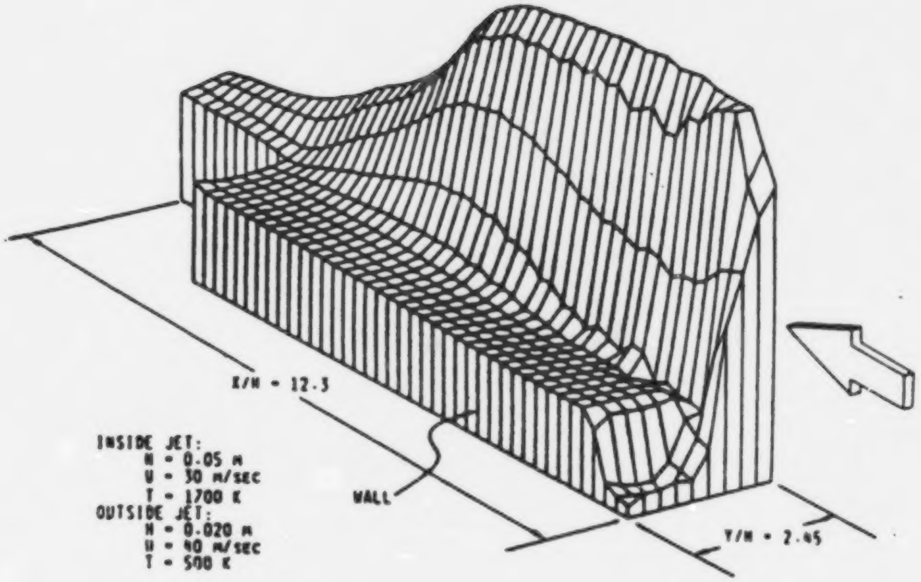


Figure 6. Contour Plot of Temperature Distribution for 2-D Coaxial Jets Expanding into a Cavity. Faster Outside Jet

The numerical simulations also serve to highlight regions of the chamber flow where design choices directly affect the control of pollutants. To reduce NO_x production, system temperatures and/or fluid transit times must be minimized. Unfortunately, lower temperatures and decreased fluid residence times also lower the CO-to-CO_2 conversion rate. With this in mind, consider the design choices presented by Cases 1 and 2. As illustrated by Figure 5 and Figure 6, the gas temperature in the major recirculation zones in both cases is between 500 and 600 K. Although this is sufficiently low to reduce additional NO_x production, it is also too low to promote further conversion of CO-to-CO_2. As illustrated in Figures 7 and 8, heat transfer rates for both processes are initially high near the injectors, nearly uniform near the wall, and substantially higher at the chamber exit. The higher momentum secondary air, however, produces much higher rates near the inlet. Since NO_x is generated in regions of high heat flux, Case 2 conditions indicate a greater potential for additional NO_x production. Furthermore, since the upstream recirculation region of Case 2 is substantially larger, and the centerline temperatures are generally lower, slightly higher outlet CO levels might be anticipated.

Preliminary Combustion Kinetics Calculation

To evaluate the accuracy and the limitations of several reaction kinetics algorithms available on the market today, preliminary comparison of methane-air combustion in a rectangular furnace using three different methods are presented. Detailed discussion of the in-depth study can be found in Mueller and Ushimaru, 1987.

For this simulation, the CPS (Combustion Process Simulator) was used to calculate the chemically reacting flow. Twelve equations for tracking the reactions of primary combustion products and intermediates plus six equations for NO_x formation were simultaneously solved. The computational problem definition is given in Figure 9. The orthogonal grid has 26 uniform nodes in the vertical direction and 92 nodes in the horizontal direction, with most nodes concentrated near the site of injection for better resolution. The three algorithms tried in this study are: (1) the chemical equilibrium (EQI) solution which assumes infinite rate kinetics, (2) the Perfectly Stirred Reactor (EQI) model which assumes infinite rate mixing between fuel, air, and intermediates, and (3) the CPS code based on finite rate reaction and finite rate mixing.

Results of this study are summarized in Figures 10 through 12. Contours of CO, OH radicals, and NO in moles/kg are shown on a log scale where every fourth contour represents a ten-fold change. The EQI solution creates and quickly burns CO in a physically unrealistic manner. The PSR and CPS still show some unburned OH radicals in this part of the burner. NO is also through to be created in the hottest regions of the burner. The EQI solutions show a large production of NO in this region. The maximum appears to be on the high side, with about 10,000 ppm at the maximum and 300 ppm at the exit plane of the burner. The PSR solution is quite different with about 300 ppm at a maximum well down stream of the hottest point and 50 ppm exiting the burner. The CPS solution is different still with a maximum of about 3000 ppm created slightly down stream of the maximum temperatures. The NO production is not highly correlated with the temperature

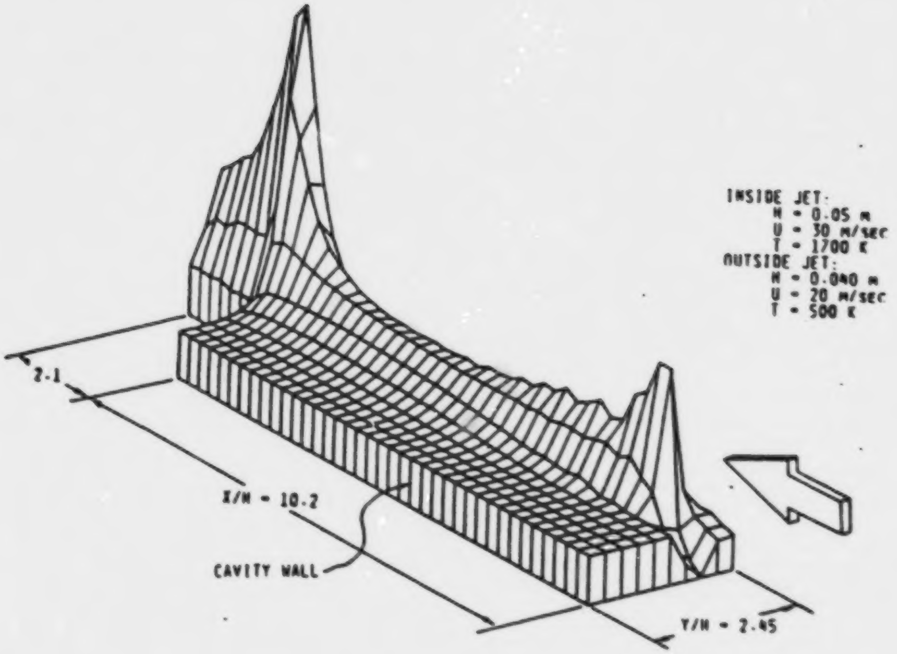


Figure 7. **Turbulent Heat Flux Distribution for 2-D Coaxial Jets Expanding into a Cavity. Slower Outside Jet**

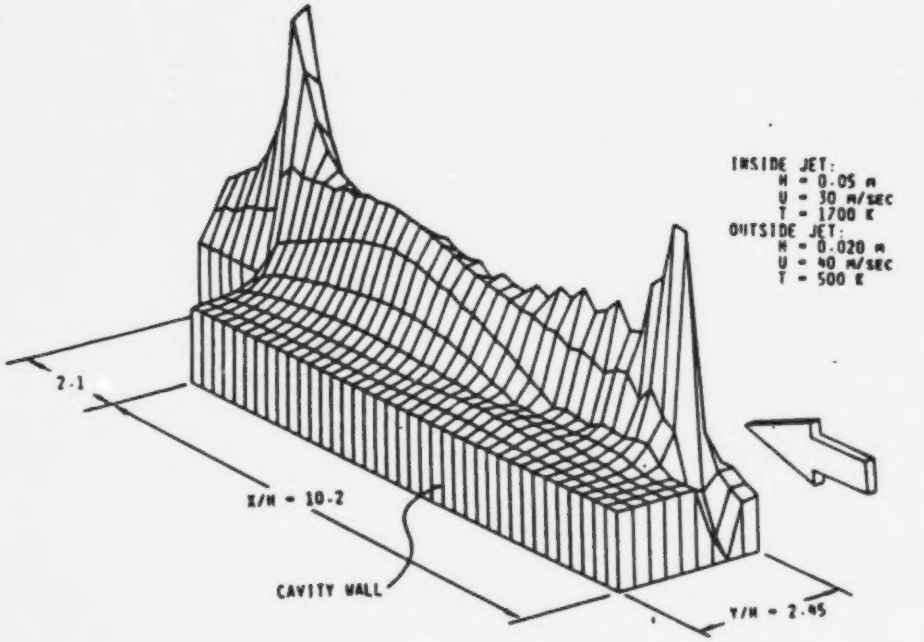


Figure 8. **Turbulent Heat Flux Distribution for 2-D Coaxial Jets Expanding into a Cavity. Faster Outside Jet**

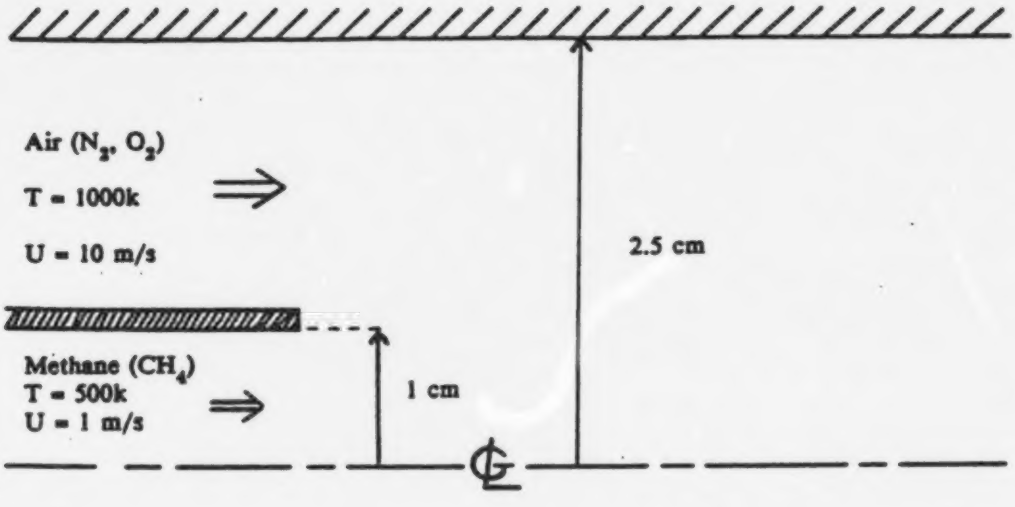

Air (N_2, O_2)
T = 1000k
U = 10 m/s
2.5 cm
Methane (CH_4)
T = 500k
U = 1 m/s
1 cm
℄

Figure 9. Methane-Air Mixing Combustion Model

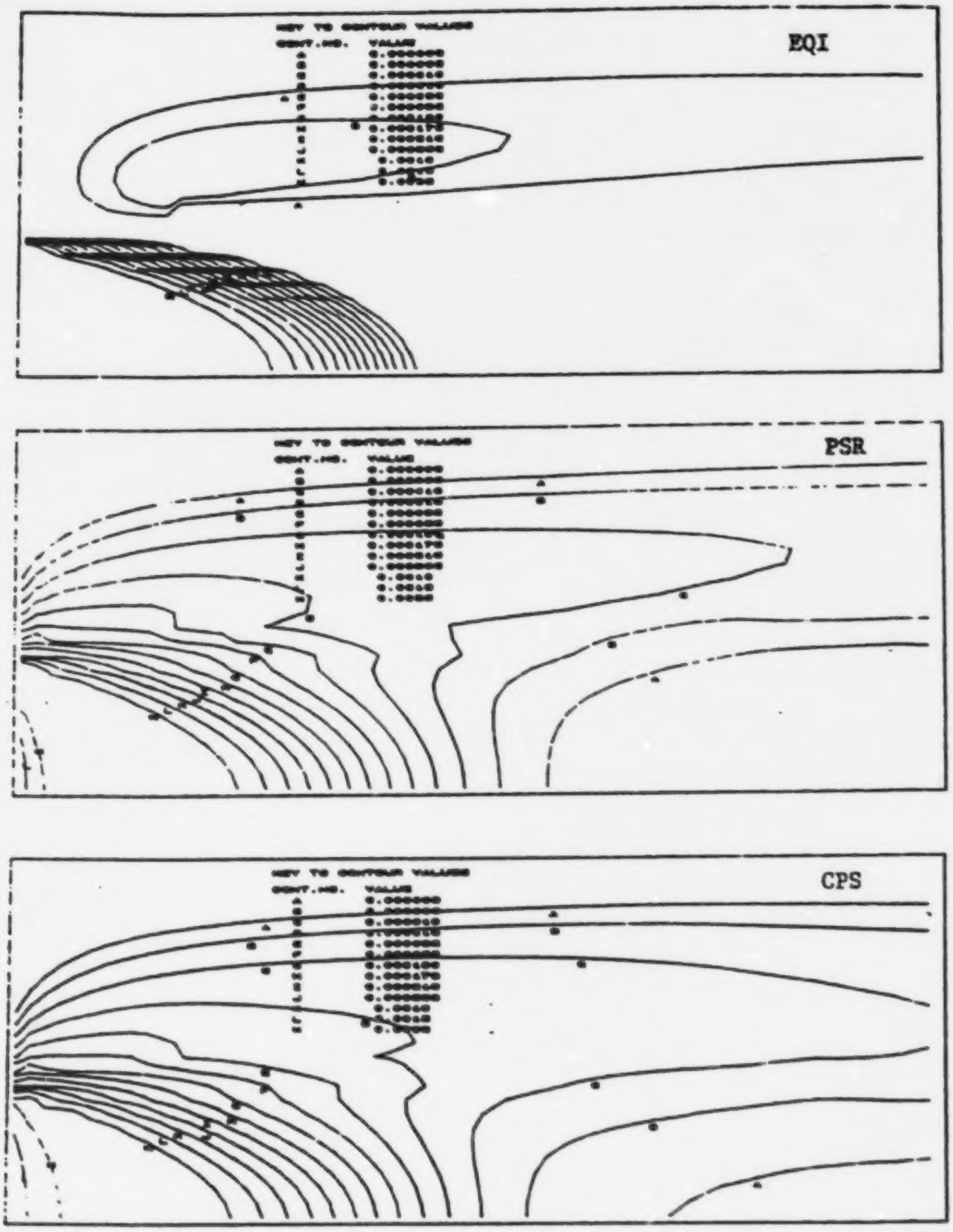


Figure 10. Carbon Monoxide (Moles/kg)

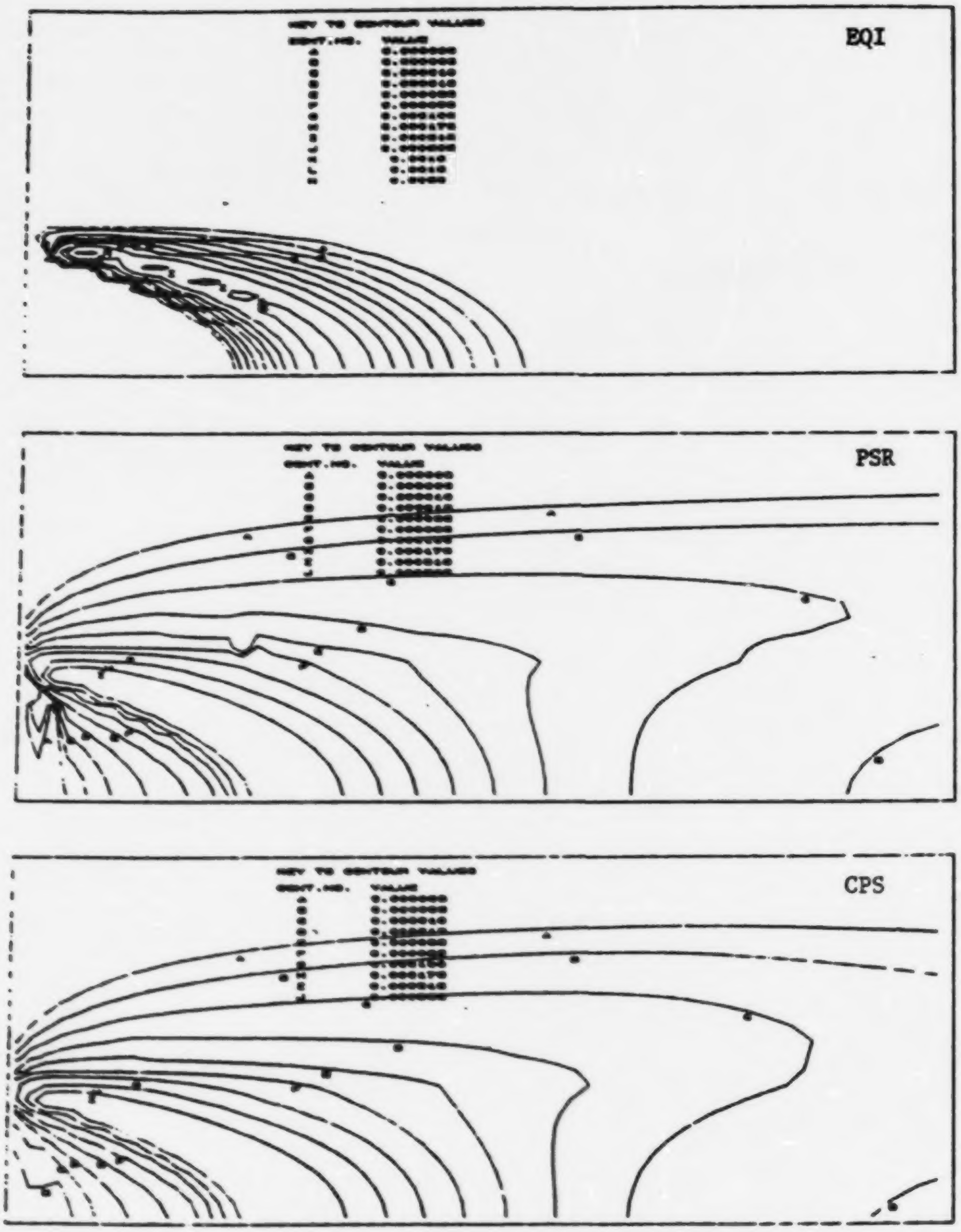


Figure 11. OH Radicals (Moles/kg)

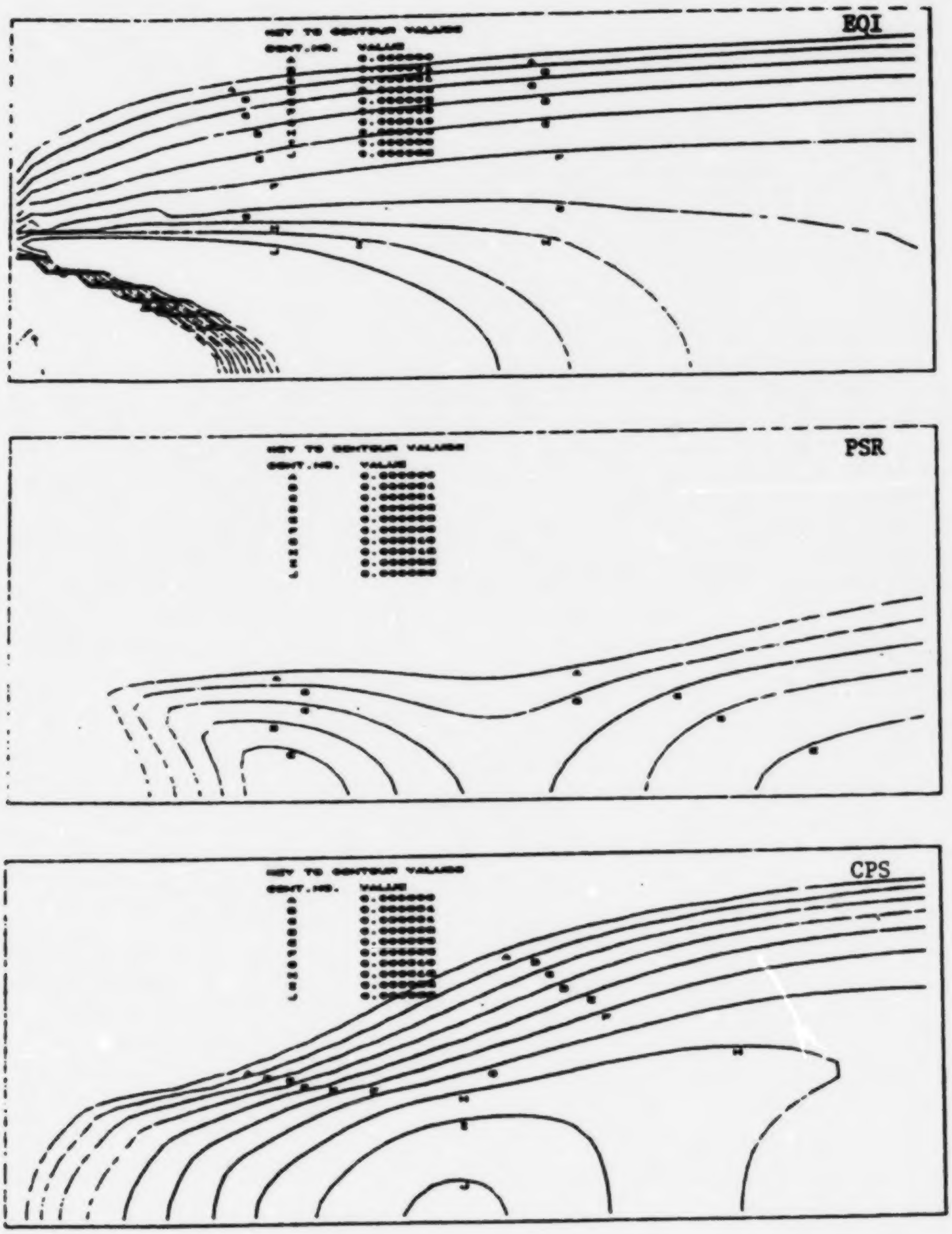


Figure 12. **Nitrous Oxide (NO) (Moles/kg)**

contours but, in fact, seems to cross the temperature contours perpendicularly. NO_2 production is also greater in the CPS model by about three times. In summary, the equilibrium model burns hottest with ignition near the point of initial contact of the fuel and air, as expected. The PSR and CPS solutions show the typical delay in ignition (due to finite rate reaction) and produce solutions which are similar in many respects. However, the PSR solution under predicts the amounts of NO_x, while the CPS solution shows a ten fold-increase in agreement with experimental data.

8. Summary

The basic elements of a design and analysis methodology for thermal processing systems have been discussed. The keystone of the approach is the PS-family of numerical simulation codes for three-dimensional turbulent flows. The technique's generalized treatment of physical geometries and associated boundary conditions, as well as its numerical stability, make it a practical engineering tool. Numerical simulations of furnace burner flow and methane-air reaction have been used in this paper to further describe the fluid dynamics, mixing, reaction, and heat transfer processes in typical industrial burners. It is clear from these examples that the codes can be used to identify potential design problems and solutions for a wide variety of thermal processing systems. Immediate applications include direct and indirect gas-fired burners, high temperature convective burners, high intensity burners, and various low NO_x burner techniques.

SECTION

IX

EXPERIENCE WITH INDUSTRIAL COGENERATION

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
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by Donald W. Petersen
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Introduction

Good morning. It's a pleasure to participate in this conference and I welcome the opportunity to discuss PG&E's perspective on the topic of cogeneration and utilities. Cogenerators and other alternative energy producers will provide a very significant part of PG&E's resource mix. In fact, as I will show you during my presentation, it is estimated that these non-utility producers will be providing over 45 percent of the additions to PG&E's resource mix by the late 1990s.

In order to integrate these non-utility resources successfully into our resource mix, more flexibility is needed to ensure that the timing, pricing and capacity of these resources reflect the need for power. This flexibility will help to ensure that the utility's customers do not pay for energy and capacity they do not need, nor to pay artificially higher prices for energy and capacity. This in turn will help us to compete effectively to retain customers that would have otherwise considered bypassing the utility system.

Public Utilities Regulatory Policies Act (PURPA) and Its Impact on Cogeneration

Technological advances, pricing structures that are not cost-based and deregulation of gas markets have given some industrial customers enough incentive to bypass or consider bypassing PG&E's system completely. Later in my presentation I will discuss PG&E's efforts to retain these industrial customers on our grid. At this point I would like to discuss another incentive to encourage cogeneration and small power production: PURPA. PURPA was signed into law by President Carter in 1978. During the initial implementation of PURPA, regulators and other utilities assumed that cogenerators and small power producers, Qualifying Facilities (QFs), would make modest contributions to the resource picture. With this assumption in mind, everyone did their best to encourage QF development. In California, the California Public Utility Commission (CPUC), in a joint effort with utilities and QFs, developed standard offers.

The standard offers achieved their purpose of encouraging QFs and then some. They included payments for both capacity and energy. The fixed energy prices contained in Standard Offer Number 4, were set in 1983 when oil prices were high and energy price forecasts were escalating. Capacity prices were also fixed and did not adjust as more capacity was added to the system. In other words, QFs were offered a great deal. Basically, the value of a Standard Offer 4 is 6-8 cents per kwh while the cost of an average cogeneration facility is 5 cents per kwh.

A burgeoning non-utility business sector will now be providing a major portion of PG&E's future generating capacity. At PG&E, QFs have signed contracts for a total of 9421 mw (of which 3519 mw are gas-fired cogeneration). Already, 2464 mw are operational and PG&E forecasts that another 2676 mw are likely to be available. As you can see from the next slide, most of these projects are cogeneration.

Some QF projects choose to use the power generated on-site for their own use and sell the surplus to PG&E. These projects added to PG&E customers that have bypassed PG&E's grid completely by installing self-generating facilities, represented a bypass load of about 700 mw during the fourth quarter of 1987.

Because of incorrect forecasts, such as those for inflation and oil prices, an overpayment problem to QFs has developed. Oil prices forecasted in 1980 deviate from 1987 actuals by 189 percent. Inflation rates were incorrectly forecasted and when used to create levelized prices led to significant front-end loading. The overpayment problem is compounded by the fact that many of the signed contracts are fixed and therefore inflexible to the changes in the utility market place.

PG&E estimates that it will be paying several hundred million dollars in overpayments by 1990. However, PG&E is not the only utility facing an oversupply and overpayment problem. Among the utilities that will probably be hardest hit are Houston Light and Power that estimates it will be paying $500 million of overpayments over the next eight years and Niagara Mohawk at $180 million over the next fifteen years.

State and Federal regulators have acknowledged these inefficiencies created by PURPA and are considering how to allow competitive forces to create a more efficient energy market. For example, the CPUC has adopted a program whereby if the utility identifies a need for a baseload or intermediate load resource, bids from QFs will be solicited. If the offer is oversubscribed, a second price auction will be implemented to determine the price to be paid winning QFs. If oversubscription does not occur, QFs will be paid the utility cost to build and operate the resource. PG&E supports the CPUC's efforts to implement a competitive bidding system.

At the Federal level, FERC recently issued three proposed rules for comment. The rules propose to: (1) refine avoided cost pricing and the restrictions in the current PURPA rules, (2) develop a voluntary bidding system, and (3) relax the regulatory requirements under the Federal Power Act for Independent Power Producers (IPPs). (IPPs are non-QF entities that sell electric power in areas where they have no service franchise and otherwise lack significant market power.)

PG&E's Cogeneration Strategy

PG&E plans to address the oversupply and overpayment problem through a deferral program. PG&E's goal is to defer viable QF projects until as close as possible to the time we expect to need the power. By "viable" we mean the QF can demonstrate it could have met its 5 year operating deadline included in its contract.

We have already initiated discussions with some QFs to defer their projects. PG&E has informally discussed its deferral program with the CPUC and the California Energy Commission (CEC) staff. The staffs have responded favorably toward PG&E's efforts. The deferral process benefits ratepayers by reducing the problems of timing, pricing and quantity of QF resources. QFs may realize savings by not rushing construction to meet their 5 year operating deadline.

Now, turning our attention to customers who are currently considering cogeneration or bypass, PG&E is implementing an aggressive marketing program to convince them to stay with the utility instead. In 1987, PG&E persuaded customers to defer 205 mw of planned cogeneration projects. In addition, PG&E also won regulatory approval for six negotiated electric rate agreements that retained 65 mw of load. Late in 1987 we signed another negotiated rate agreement that will defer the construction of a 100 mw cogeneration facility.

Sales figures for 1987 reflected our marketing success. Total electric sales were up 7.6 percent and gas deliveries were up 31 percent.

The Future of Cogeneration in California

As shown by this last slide, cogeneration will be a major resource addition at PG&E. It is an exciting challenge to regulators and PG&E to ensure that cogeneration becomes an efficient and competitive part of our resource mix.

Additionally, PG&E will continue its aggressive marketing program to retain customers who are considering cogeneration. Thanks to our redoubled efforts to cut costs and market our products more competitively, PG&E has already managed to reduce by 50 percent the substantial revenues we once estimated could be at risk by 1990. Our job isn't finished as we would like to be the supplier of choice to as many customers as possible. We will continue to emphasize the value we can provide our customers through competitive rates and quality service.

SECTION

X

FUEL CELL DEVELOPMENTS

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Mike Papic
B. C. Hydro

9/5/666

ABOUT THE SPEAKER:

M. M. Papic received his B.Eng. and M.Sc. degrees from the University of Zagreb and his Ph.D. degree from the University of Ottawa. In his professional career, he has been involved in studies of chemical kinetics, heat and mass transfer operations, thermal processes for coal and solid waste conversion, and environmental problems, including air pollution, water treatment and solid waste management. Dr. Papic is the Supervisor, Electrochemistry Section, B. C. Hydro Research and Development. He is the Manager of the Fuel Cell Power Plant Testing project.

Dr. Papic has published a number of papers in the field of chemical engineering, chemistry and education.

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"Big deal. Fire gives light and heat."

NATURAL GAS FUEL CELL

A fuel cell is a device in which the chemical energy of reactants (fuel and air) is converted directly into useable energy in the form of electricity and heat without combustion as an intermediate step.

Fuel cell power plants are distinguished by high efficiency, low emissions, fuel flexibility, quick installation and quiet operation.

The process concept was discovered in 1839 by Sir William Grove a British scientist, and developed into a useable technology in the nineteen sixties by NASA scientists and engineers.

The process concept is similar to that used in a common primary electrochemical cell and is the reverse of the water electrolysis process.

A single fuel cell is composed of an anode (an electrode where oxidation occurs) and a cathode (an electrode where reduction occurs) separated by an electrolyte. The electrodes are porous plates sandwiched around porous material holding the electrolyte.

A single fuel cell generates dc current at about $2kA/m^2$ at a potential of about 1 V. Practical voltages are obtained by connecting many individual cells in series i.e. stacks or batteries (456 cells, 225 kW). Fuel cell stacks can be configured in series, parallel, both series and parallel or as single units, depending on particular application.

For a fuel cell power plant to be a practical energy generator it should be able to use commercially available fuels and be able to produce alternating current (ac) power. Thus a fuel cell power plant consists of three major subsystems; a fuel processor, a fuel cell power section and a power conditioner.

Fuel cells have been developed for a number of diverse applications including space power, naval propulsion, electric vehicles, isolated power sources, prosthetic applications and utility applications.

Utility applications include servicing homes, apartments, institutions, commercial establishments, industrial installations, power distribution sites, isolated generating plants, central generating plants and peak generating plants.

Integrated into cogenerating energy packages fuel cells can supply electricity, heat and air conditioning services.

The total mean electricity demand on the B.C. Hydro system which could potentially be supplied by fuel cells is 909 MW. If all this electricity was generated by natural gas fuel cell at 40% efficiency then the natural gas demand would amount to 71.7 PJ/year which is about 72% of total B.C. Hydro natural gas sales.

A simplified economic evaluation of fuel cell power plants based on a project installed cost of between $1,000/KW and $2,000/KW (1985 Canadian Dollars), with an assumed 30-year life, a 6 percent (net of inflation) interest rate and natural gas fuel available at $3.08/GJ ($3.25/MMBtu) indicates the cost of power ranges from 6¢/KW.h to 9¢/kW.h depending on the size of the plant, actual installed cost, anticipated operating and maintenance cost, and projected heat rate. No allowance has been included for use of the thermal energy available from the power plant. Full utilization of the thermal energy would effectively reduce the cost of power by 50 percent. This compares well with the cost of alternative energy systems.

AL. FUMES
CHEMICAL
CO.

"Sorry George . . . looks too expensive for us."

FUEL CELL TESTING PROGRAM AT B.C. HYDRO

Before fuel cell equipment can be introduced into a utility system, appropriate technical information, performance data and operating experience is required to determine the effect on a utility system operation and to develop an acceptable level of confidence. Thus an operational feasibility program has been undertaken to establish the acceptance of and commitment to fuel cell energy systems for utilities.

B.C. Hydro has installed an on-site 40 kW phosphoric acid fuel cell (PAFC) power plant in a road mobile trailer. The experience gained in installing and operating this natural gas-fuelled unit will be invaluable in specifying the general performance requirements for Canadian utilities. The installation constitutes a natural gas fuelled mobile fuel cell power system demonstration unit complete with controls and monitoring equipment and is the only one of this kind in Canada. The test project provides a unique opportunity for participating utilities to share in the operation and testing of a fuel cell power plant and in the development of a fuel cell generation program for Canada.

B.C. Hydro and Ontario Hydro jointly supported the Fuel Cell Test Program at the B.C. Hydro Research and Development Centre.

(a) Grid - Independent Mode of Operation at B.C. Hydro R&D Centre

The operation and testing of the power plant in the grid-independent operating mode was conducted at the R&D Centre from November 1985 to September 1986. During that time a total of 5253.6 operating hours were accumulated. The longest continuous run was 2205.6 hours which is the longest continuous operation recorded with this type of fuel cell power plant. The availability of the power plant during the period was 68% which compares well with the cumulative availability of forty six power plants employed in the GRI/DOE program of 63%.

There were no failures of the unique components: the fuel cell stack or fuel processor. The failures that occurred were primarily with mechanical pumps and valves and electronic logic components.

(b) Grid-Connected Mode of Operation at BCH R&D Centre

During October 1986 the power plant was shut down to allow the installation of the grid-connect unit (GCU). A thorough inspection of the power plant was made; a new water pump was installed; and a number of repairs and modifications were made to the feed water module and heat management system.

The installation of the GCU was completed 27 October and the unit was recommissioned 28 - 31 October 1986 in the presence of the International Fuel Cells field engineer. The power plant operated in grid-connected mode from 3 November 1986 to 8 May 1987. During that time a total of 2869 operating hours were accumulated. The longest continuous run was 1452 hours. The availability of the power plant during that period was 64%. The best recorded 30 minutes electrical generating efficiency of 40.4% and the best 30 minutes thermal recovery of 39.4%, i.e. the overall energy efficiency of 79.8%.

(c) Demonstration at the Sungod Aquatic Centre

Innovations in energy generation alternatives are an essential part of B.C. Hydro's future. But such innovations will have to be demonstrated in real-life situations if they are to be seen as commercially viable and accepted by consumers.

This project has been undertaken to install the fuel cell power plant in a commercial application at the Sungod Aquatic Centre, Delta, B.C. and operate it for at least six months duration.

This project will allow monitoring of the performance of the fuel cell power plant in an actual commercial application. We will be able to develop a technology transfer plan with detailed technical documentation, engineering and installation data, operating results, project costs, staff requirements, and environmental considerations. The project will also raise public awareness of this new technology.

The estimated cost of the project (six months) is $150,000. Following are the contributors for the project:

Energy, Mines and Resources Canada
B.C. Hydro
Canadian Western Natural Gas
Consumers Gas
Gaz Metro
Greater Winnipeg Gas
ICG Ontario
Inland Natural Gas
Northwestern Utilities
Ontario Hydro
Pacific Northern Gas
Union Gas
West Coast Transmission

SECTION

XI

PACKAGED COGENERATION SYSTEMS

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B.C.
13 - 14 April, 1988

Gary Warneboldt
Finning Ltd.

ABOUT THE SPEAKER:

Gary Warneboldt's career with Finning Ltd., dealer for Caterpillar engines, began in November of 1966. Gary's many years of experience in various positions such as Journeyman Heavy Duty Mechanic and Journeyman Diesel Electrician, Technician Specialist, Service Sales Coordinator, Branch Coordinator and Engine Sales Representative has given him a thorough background in the diesel power generation field.

Finning assembles and sells Generator Sets from 50 kW to 4500 kW. They also manufacture Switchgear and Heat Recovery Systems.

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SUMMARY NOTES

The following are summary notes only of Gary Warnebolt's presentation, for more detailed information, contact Gary directly.

PACKAGED COGENERATION SYSTEMS

Slide 1 Cogeneration

Cogeneration is the simultaneous production of electric and/or mechanical power and useful thermal energy by utilizing internal combustion engines or turbines.

Slide 2 Adverse forces acting upon utilities

- increasing regulation
- environmental activism
- consumer pressure against rate increases
- new plant capital requirements

In 1976, it was planned that an additional 129 nuclear plants and 218 coal fired plants with a combined capacity of 260,862 MW would be on line in North America between 1986 and 1989. A very large percentage of these have been delayed, postponed or cancelled altogether.

Slide 3

We in Western Canada and the U.S. Northwest have been sheltered somewhat from these adverse factors due to our limitless hydro electric resources. Is this going to change? Careful application of packaged cogeneration systems may soon be seen here as a cost effective supplement for selected energy users.

Slide 4

There are two basic approaches to cogeneration:

- bottoming, and
- topping cycles.

A bottoming cycle is one in which the fuel input is first applied to a useful thermal energy process. Rejected heat from this process is then used for power production. Steam turbines operating on otherwise wasted heat would be an example of a bottoming cycle.

Slide 5

A topping cycle installation uses a reciprocating internal combustion engine or turbine in which the fuel is first used to produce useful power output. The rejected heat from this process is recovered and used for heating or cooling purposes. Where does this rejected heat come from ?

Slide 6 Typical Breakdown of Fuel Burned in a Reciprocating Engine

Typically 33% of the fuel input is used to produce mechanical power while 67 % generates usually wasted heat.

Slide 7

The most convenient way to recover waste heat is by means of a water jacket. Fuel energy utilization can be increased to the 60 - 65 % range.

Slide 8

Approximately 1/2 of the heat rejected is recoverable economically through heat recovery silencers.

The danger of overcooling has to be considered.

This will improve system efficiency by another 12 - 15 % which will bring the total system thermal efficiency to the 70 - 75 % range.

Slide 10

Typical applications for heat recovery are: domestic hot water supply, laundries, washing units, boiler feed preheating, swimming pool heating, process heating for liquids other than water, and combined steam systems with absorption air conditioning.

Slide 11 Benefits of Cogeneration

1/ Cogeneration reduces the total energy costs by generating electricity (or mechanical power) and useable heat in a more cost effective manner than by their separate generation.

2/ Cogeneration provides an alternate power source in the unlikely event of an utility outage, possibly eliminating the need for separate standby generators and transfer mechanisms.

Slide 16

The thermal electrical load profile is important with respect to the sizing of a cogeneration system. It should have the capacity to meet the base load requirements of the thermal and electrical demand. If this is done the potential waste of thermal or electrical energy is reduced and the cogeneration system can run as many hours as possible.

Slide 17

A sophisticated computer program has been developed for sizing a cogeneration system. The output is a discounted cash flow analysis based upon the costs discussed already, plus federal and provincial tax rates, depreciation methods, investment tax credits, borrowed capital and a variety of other factors.

Slide 22

Many packaged systems include the design and supply of: engine/generator, switchgear and controls, and IT/ requirement.

Slide 23 Efficient Energy Management

In summary, there is a significant and growing opportunity for energy management. Those of us involved in the energy sector are facing some unusual times. Times which will call for careful consideration of all potential energy sources.

The main purpose of any cogeneration or peak shaving system lies in meeting the needs of both the user and the utility.

SECTION

XII

TURNING PRESSURE INTO ENERGY

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Anthony Cleveland
REACT Energy Ltd.

5/666

ABOUT THE SPEAKER:

Tony Cleveland graduated in England from the University of London with a B.Sc. degree in Mechanical Engineering.

Tony has had some 35 years experience with major organizations in England, the USA and Canada in engineering design, development, procurement and management in oil and gas installations and nuclear power plants. He is a specialist on gas turbines, compressors, heat exchangers and control systems.

Tony has been working with the oil and gas industries in Alberta since 1975, and is new General Manager of REACT Energy Ltd., a Calgary-based company established in 1985 to engineer and manufacture energy recovery systems. The company manufactures turbo-expanders to generate electricity by recovering the energy normally wasted in gas pressure letdown applications.

TransAlta Energy Systems is a Canadian company dedicated to the application of the latest technology to the efficient use of energy. In addition to the React Turbo Expander, the company offers a full range of services for building automation.

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9/5/652

TURNING PRESSURE INTO ENERGY

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ABSTRACT

In industrial and commercial installations, there are many instances where energy is wasted, or at least under-utilized.

In many cases it may be possible to recover some of this energy and obtain useful work, thereby improving plant efficiency and the economics of the operation.

The turbo expander is a simple device that can make a significant contribution to the recovery of energy in many kinds of plants.

This paper considers some ways in which turbo expanders may be used, and looks in detail at an application in the gas industry where the energy lost in pressure reduction may be recovered and operating costs reduced. The design criteria for such turbo expanders are discussed and areas for future development are proposed.

This paper discusses specific installations where pressure letdown energy is being recovered by using a proven turbo expander driving an electrical generator. The advantages of this system, which is commercially available from a Canadian company, are outlined.

The paper concludes that there are significant gains to be made in the recovery of waste energy and that the turbo expander can plan a major role in this activity.

INTRODUCTION

The oil and gas industry is not simply a producer of energy, it is also a large consumer of energy. For example, heavy oil recovery involves injecting gas, steam or other fluids under high pressure in order to recover the maximum quantity of oil from the reservoir. This process requires large amounts of energy expenditure to produce the high injection pressures.

In addition, the energy which is available from a hydrocarbon source is sometimes under-utilized, if not actually wasted. Examples of this can be seen in gas which is flared, heat energy which is allowed to go to waste, and pressure energy which is lost without producing useful work.

In some cases this loss of energy is unavoidable. For example, low grade heat cannot always effectively be utilized. However, it may be possible to recover some of this energy and realize useful work, thereby improving the economics of the operation. One possible method of energy recovery is to turn wasted pressure letdown into energy using a turbo expander.

In processing natural gas, a turbo expander can be used to produce the cooling required to separate the hydrocarbon liquids from the gas. The power produced by the turbo expander can then be employed to drive a compressor to compress the dry gas after liquid separation. This reduces the net energy input to the plant.

Another example, again from the gas industry, relates to the heat rejected to atmosphere in the exhaust stacks of gas engines and gas turbines. This heat, which may represent over 60% of the energy in the gas supplied to the engine, is a significant loss which adds substantially to the operating costs. Some of this energy may be recovered and put to good use. The waste heat in the exhaust can be recovered using a Rankine cycle in which a working fluid, such as steam or ammonia, may be used to drive an expansion turbine. This process was successfully employed in a compressor station for Nova in Canada, and increased the overall thermal efficiency of the plant from 34% to 43% (Reference 1 & 2).

The turbo expander is simply a machine in which fluid or gas at high pressure is expanded to a lower pressure and thereby does mechanical work. A common example of a turbo expander is the turbo charger on a gasoline or diesel engine, where exhaust gas is expanded through a turbine and the energy derived is used to drive a compressor to boost the engine charging pressure, thereby increasing power and efficiency.

There are many other such examples of the power of turbo expansion. The thermodynamic principles are well known but outside the scope of this paper. Performance depends upon the fluid being used, the inlet and outlet conditions of temperature and pressure, and the efficiency of the process. All these factors are known or may be determined in each particular case.

The turbo expander has an important part to play in the recovery of energy. A turbo expander can be used to recover energy lost in pressure reduction, to recover energy from waste heat, and to provide refrigeration and cooling. Recovery of energy from waste heat includes low grade heat from exhaust steam, heat from geothermal sources and, as already mentioned, heat rejected from gas turbines and reciprocating engines. The refrigeration and cooling properties of a turbo expander can be utilized in gas processing, food processing and air separation, as well as in air conditioning for hospitals, offices and shopping centres. In this paper, we shall be concentrating on the use of turbo expanders to recover energy from gas pressure reduction. Specifically, this paper will discuss installations where pressure letdown energy is currently being recovered in a system which couples a turbo expander of proven design to an electrical generator. It will also outline the advantages of this commercially available system.

RECOVERY OF ENERGY IN PRESSURE REDUCTION

The particular case of energy recovery in gas transmission and distribution has been selected to illustrate the use of a turbo expander.

In order to provide efficient transportation in pipelines, gas has to be pumped at high pressure. However, the pressure has to be reduced at the delivery points, and a proportion of the energy used in transmission is then lost. This energy is normally dissipated in pressure regulating valves. Figure 1 shows the typical pressure profile of a transmission system from well head to customer.

A turbo expander, operating on the pressure drop available in the gas from transmission line pressure to distribution line pressure, can be used to drive an electrical generator which can then provide power to a plant, an industrial complex, or for sale to the electric utility. The turbo expander unit is controlled automatically so that it exactly duplicates the functions of the regulator valves.

To illustrate the ways in which a turbo expander may be used to recover this pressure energy, a typical wellhead system and an installation at a sales gas station where pressure has to be reduced to the customer are shown diagrammatically in Figures 2 and 3.

While well head applications are significant from the point of view of numbers of sites involved, and the additional benefits of liquid recovery as well as electricity generation that can be provided at site, the recovery of energy at city gate stations and process plants using natural gas is more important economically, because the power liberated at any one site is more substantial and may be more readily utilized in local electric systems.

This method of pressure regulation which recovers energy otherwise lost in the process, has already been used in a few installations in North America and in Europe. One of the reasons why this technique has not been exploited more widely to date is that the cost of such installations in relation to their output has traditionally been relatively high.

The vast majority of pressure control stations on gas distribution systems do not develop on average much more than 500 kilowatts (kW). To ensure economic viability, small turbo expanders have to be available with the flexibility of operation to handle the flow variations expected, and at the same time having low capital cost per kW developed. Figure 4 illustrates the distribution of power output available from a number of pressure reduction stations in the Province of Alberta. It will be seen that a significant proportion have an output of 500 kW or less.

The commercial availability of a simple, low cost turbo expander having an output range of 100 kW to 500 kW makes the prospect of energy recovery from city gate stations a realistic proposition.

SYSTEM COMPONENTS

Confidence in a turbo expander pressure letdown energy recovery system is based upon the use of well-proven equipment. Firstly, the turbo expander is a simple device based upon established technology. Specifically designed versions have proven themselves rugged and reliable in natural gas applications throughout the world. The key to a successful installation is the selection and optimization of the design to suit the particular operating conditions.

The typical turbo expander installation consists of the turbo expander unit with its own lube and seal system, attached to a gearbox and electrical generator, and mounted on a skid. The turbo expander has its own control panel for start up and shut down, as well as a control system for monitoring operating conditions and controlling the variable flow control valve which adjusts to meet differing flow requirements (Figure 5).

The installation also includes a bypass line with a pressure control valve that can be put into operation at any time the turbo expander reaches its upper or lower flow limits, or when the expander is to be taken out of service for maintenance. In addition, block valves will be provided to shut off the turbo expander for maintenance or replacement purposes. The complete assembly, Figure 6, is a self-contained unit which can replace a pressure reduction valve and will function in exactly the same manner.

The electrical and control equipment must be carefully chosen to ensure that it is entirely compatible with the user's operational requirements. With the current availability of sophisticated and very reliable microprocessors, the control problem is greatly simplified.

THE TURBO EXPANDER UNIT

To serve the market for recovery of energy from pressure reduction, the ideal turbo expander until needs to have the following characteristics:

1. Reliability
2. Low Capital and Installation Costs
3. Low Operating Cost
4. Low Maintenance Cost
5. Wide Operating Range with Efficiency
6. Rugged and Simple Design

In many applications, item 5 in the list above is particularly important, as the operating conditions may vary widely throughout the year.

From the operator's point of view, the reliability and maintainability are important criteria. This unit is replacing a pressure reducing valve, and it is essential that it should behave in all aspects as a valve and should not impose an additional operational burden.

THE REACT TURBO EXPANDER

The React turbo expander is a simple low cost unit embodying a single stage axial flow turbine, with fixed nozzles and an integral speed reducer gearbox. This unit is mounted on a baseplate and coupled by a flexible coupling to the generator. The skid embodies its own oil reservoir and carries a lubricating pump and filter. The skid for a 125 kW React turbo expander is approximately eight feet by four feet and weights about 3000 pounds. A weatherproof acoustic cover is provided over the whole unit so that it can be installed outside in any convenient location.

The basic React expander unit is capable of a rating of up to 125 kW ; higher power units with outputs up to 500 kW will be available in the third quarter of 1988.

The React turbo expander unit may be placed upon a concrete pad, and inlet and outlet gas connections made through a simple piping arrangement with block valves to isolate the unit from the parallel regulating stream.

The design of the turbine is based upon a well proven turbine starter unit. The machined steel case is rated according to the pressure requirements of the installation.

The turbine rotor in the present 125 kW units is a two-piece design, with a titanium hoop containing the turbine blades or buckets shrunk onto an aluminum disc. The rotor is carried on a pair of anti-friction bearings and the extremity of the rotor shaft carries the sun gear of the integral epicyclic gearbox. Because of the modular design, it is simple to install and remove for overhaul, and repair is readily carried out.

The React turbo expander has been designed to operate over a range of pressure ratios from 1.5:1 to 7:1 without changes to the wheel. Within these limits of ratios the wheel will pas from 100,000 standard cubic feet per day (SCFD) to more than 20 million SCFD, depending on nozzle configuration.

Power varies with pressure ratio and flow, and the available envelope of operation can be illustrated by plotting power output against flow for the extreme limits of pressure ratios. The turbo expander can be designed to operate efficiently within the shaded area shown on Figure 7.

The React turbo expander is manufactured by React Energy Ltd., a Canadian company established in 1985 to engineer and manufacture energy recovery systems. React Energy Ltd. has on-going projects in Canada, Pakistan and Hungary.

The React turbo expander is marketed in Canada by TransAlta Energy Systems, a Canadian company which is in the business of applying the latest technologies to the efficient use of energy.

TURBO EXPANDER SYSTEM ENERGY BALANCE

A turbo expander utilizes the internal energy of the fluid to do work and in the process produces a corresponding reduction in temperature. Provided the temperature reduction is needed in the process, then the power available represents a net gain. This is the case in the typical gas liquids stripping application. However, if there are constraints on temperature at discharge, as for example in the process of pressure reduction at a city gate where the gas has to be kept out of the hydrate zone or to prevent ground freezing, then some additional energy has to be supplied in the form of pre-heat.

It is fairly simple to determine the net energy gain by examination of the pressure-enthalpy diagrams for the gases concerned. It soon becomes clear that in most cases, the energy made available by the process of the pressure reduction appears to be equal to little more than the energy that is required to pre-heat the gas to avoid the discharge temperature dropping below the hydrate limit. This makes the process, at least theoretically, unattractive. However, it must be remembered that a pressure reduction valve operating under similar circumstances also requires heat input to avoid this temperature limitation. Therefore, the energy balance should be amended and this amount of heat subtracted from the heat required by the turbo expander.

If this amendment is made, it will be seen that although the heat input required by the turbo expander is usually higher than the heat required by the pressure reduction valve operating under similar circumstances, there is a net energy balance in favour of the turbo expander (Figure 8).

In most cases, 15% or more of the total energy available in the pressure reduction process can be extracted and turned to good account.

The thermodynamic considerations are not however the only elements in the economics of using a turbo expander in place of a pressure reducing valve. The real benefit is that there is a positive and not a negative energy balance; that is, energy is recovered, not lost in the process, and depending on the relative costs of preheating the gas and the costs of electrical power, this energy gain can be multiplied into a significant economic benefit.

The other factor which has to be considered in evaluating the economic benefits available from a turbo expander is the cost of the installation. An expander costs more than a valve but can contribute to revenue and realize significant economic benefits. The capital cost of the installation has to be paid for out of the revenue obtained; therefore, to ensure a reasonable payback period, the capital cost has to be kept within reasonable limits.

The decision to use a simple design, single stage turbine, designed as part of a modular package, is the key to the control of capital cost.

By using a unique design of turbine wheel, a wide range of flow and pressure ratio conditions can be accommodated without major changes to the module.

ECONOMICS

The economic benefits of using a turbo expander for pressure regulation depends upon the power available, the cost of power, the cost of preheat, the utilization factor and the capital cost.

Of these, the cost of power and the utilization are of major importance. The installed cost has to be weighed against the revenue obtained in order to evaluate the payback. To maximize revenues, the hours per year of operation should be high and the rate applicable to electric power should also be high. Using typical installation costs and a utilization of 95%, electric rates of 3 cents per kilowatt-hour or more should provide an economic rate of return and a payback of less than three years.

For example, if gas is used for preheating at say $2.00 per million BTU, and electricity is worth say three cents per kilowatt-hour, the ratio of electric power to gas cost is approximately 4.4:1. In this case, the energy liberated by the turbo expander is worth many times more than the additional heat used in the preheaters. Therefore, the gain in energy in using a turbo expander in place of a valve to reduce pressure is multiplied by the ratio of energy costs to obtain the real economic benefit. Now if preheat energy is readily available in the form of existing waste heat, the benefits are even greater, as this preheat cost may be very low or negligible.

DEMONSTRATION UNIT

In 1987, Canadian Western Natural Gas Company Limited decided to evaluate the potential of turbo expander energy recovery at a site in Calgary.

In the selection of the station for the demonstration program the following significant parameters were taken into account:

1. The station should have an output capability to load the turbo expander to the fullest extent throughout the year.
2. The flow variation throughout the year and on a daily basis should preferably follow a consistent pattern and not be subject to major interruptions which would affect the utilization of the expander.
3. The preheating capacity of the line heater of the station should be sufficient to cater for the increased temperature drop that would be experienced through the turbo expander.
4. The site should be capable of accepting the turbo expander unit without major modifications and excessive capital expenditure.
5. There should be convenient access to the electrical power lines, again without excessive cost to provide the necessary tie in.

The results of the province wide survey of regulating stations shown on Figure 4 demonstrates the distribution of power output, and show that the vast majority of stations lie below 500 kW in average output. Average output is defined in this case by the arithmetical average of throughput and pressures throughout the year. For most regulating stations, the discharge pressure is maintained at a constant level and the input pressure to the station fluctuates within a fairly narrow band as the line pack in the system varies. In Alberta, throughput is highest in the winter, and lowest in the summer, and on a daily basis there is a variation which reflects load changes through the working hours of the day. The average output therefore tend to be somewhat misleading in terms of total potential because peaks in the winter may be many times higher than the average.

The month by month variation does not of course provide a complete picture of the capability of a station as the average monthly output is made up of a series of daily variations, which can be shown in three dimensional form in the diagram Figure 9.

From the analysis of monthly and daily flow variations at several stations in the Calgary area, together with consideration of the other parameters of physical location and facilities at each site, a final selection was made.

The site selected was at Simons Valley, in the north of Calgary; a station which has a potential of over 750 kW and an average of at least 250 kW for ten months in the year. For this installation, a 250 kW unit was chosen, consisting of two 125 kW turbines coupled at each end of a 250 kW generator. The compact nature of the unit can be seen from the photographs of the Simons Valley 125 kW installation. This twin unit, the first of its type, represents a step between the 125 kW single unit and the new 500 kW updated single unit. This new design has a slightly larger turbine wheel and a new epicyclic gearbox.

With this new unit and the modular concept, the range of coverage for pressure let down stations is complete from the smaller installations of 50 to 125 kW up to the larger city gates and industrial complexes of 500 kW and more.

CONCLUSION

There is a large opportunity for recovering wasted pressure letdown energy economically, utilizing a turbo expander system of proven design, coupled to a generator to produce electricity.

This paper has examined the use of a turbo expander as a device for recovering energy in relation to certain specific applications in the oil and gas industry. It has been concluded that it is eminently practical to use a turbo expander to generate electricity in the process of pressure reduction, and that utility companies and customers can benefit significantly from the power produced. To permit greater use of this concept in the future, particularly in applications such as well heads and smaller city gate stations, cheaper, integral high speed units may have to be produced.

At the same time, utilities, city administrations and politicians will have to take a wider view of energy conservation in terms of the public benefit in order to encourage this form of cogeneration, and allow it to make its mark as a contributor to energy conservation. A significant step in this direction has recently been taken in the Province of Alberta, with the release of the Small Power Inquiry report. The report recommends that the Alberta Government adopt a policy that would facilitate the production of electric energy by independent producers in parallel with the Alberta interconnected system; bases the prices paid to small power producers by utilities on the long term avoided costs of new generation capacity; and exempts small power producers from the full regulatory requirements of the Public Utilities Board and the Energy Resources Conservation Board. These recommendations are expected to significantly encourage the installation of small power production facilities in Alberta.

The potential for waste energy recovery and electricity production is considerable, particularly where electricity is relatively costly and where energy for pre-heating is available at low cost.

The turbo expander may also be invaluable where relatively small amounts of power are required but no convenient source of electricity is at hand. These cases are typically field installations where pressure energy is currently being wasted and electricity supplied by diesel generators using high cost fuel transported long distances by road or air.

The technique demonstrated here could be applied to a number of other pressure reduction stations and generate an appreciable amount of additional revenue. The viability of these stations, of course, depends upon the current rates for electricity, but work to date has already demonstrated that with rates in excess of three cents per kilowatt-hour, turbo expander installations of less than 500 kW can be made economically viable.

While this paper has concentrated on the turbo expander in the process of pressure reduction, and in particular the example of a city gate station, it must be remembered that this concept of energy recovery is applicable to a wide range of circumstances, including high pressure steam, exhaust gas from fire flood projects in the oil industry and of course in secondary cycles to an even wider range of applications generally in industry and commerce.

REFERENCES

1. Vogt, G.M. - NOVA Corporation of Alberta. Operating experience with a waste heat recovery system installed in a pipeline compressor station. ASME 83 GT226.

2. Malewski, W.F. and G.M. Holldorf - Borsig GmbH, Berlin. Combined cycles for pipeline compressor drivers using heat. ASME Conference, 1979

Natural Gas Transmission System

Pressure Distribution Diagram

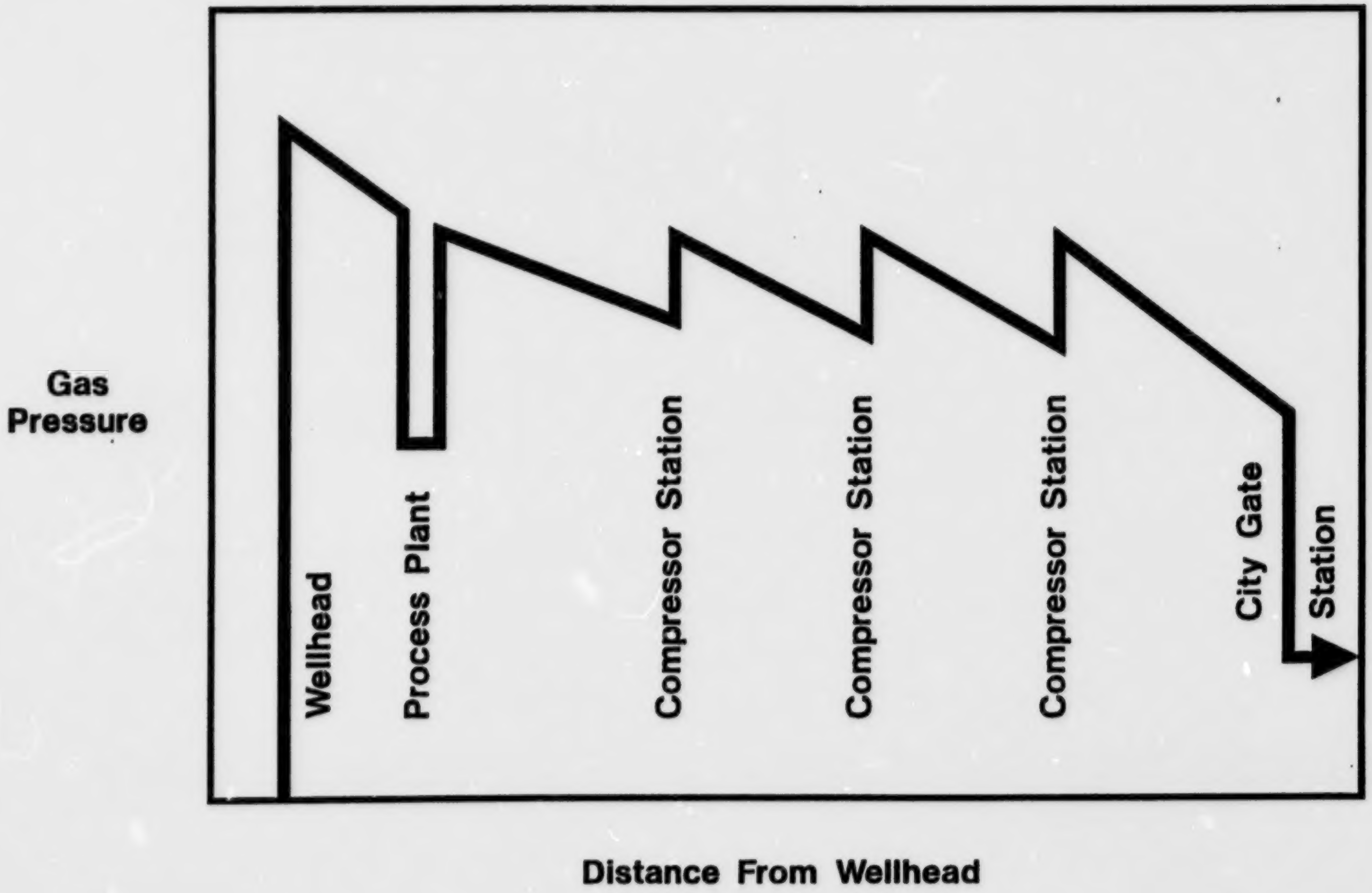
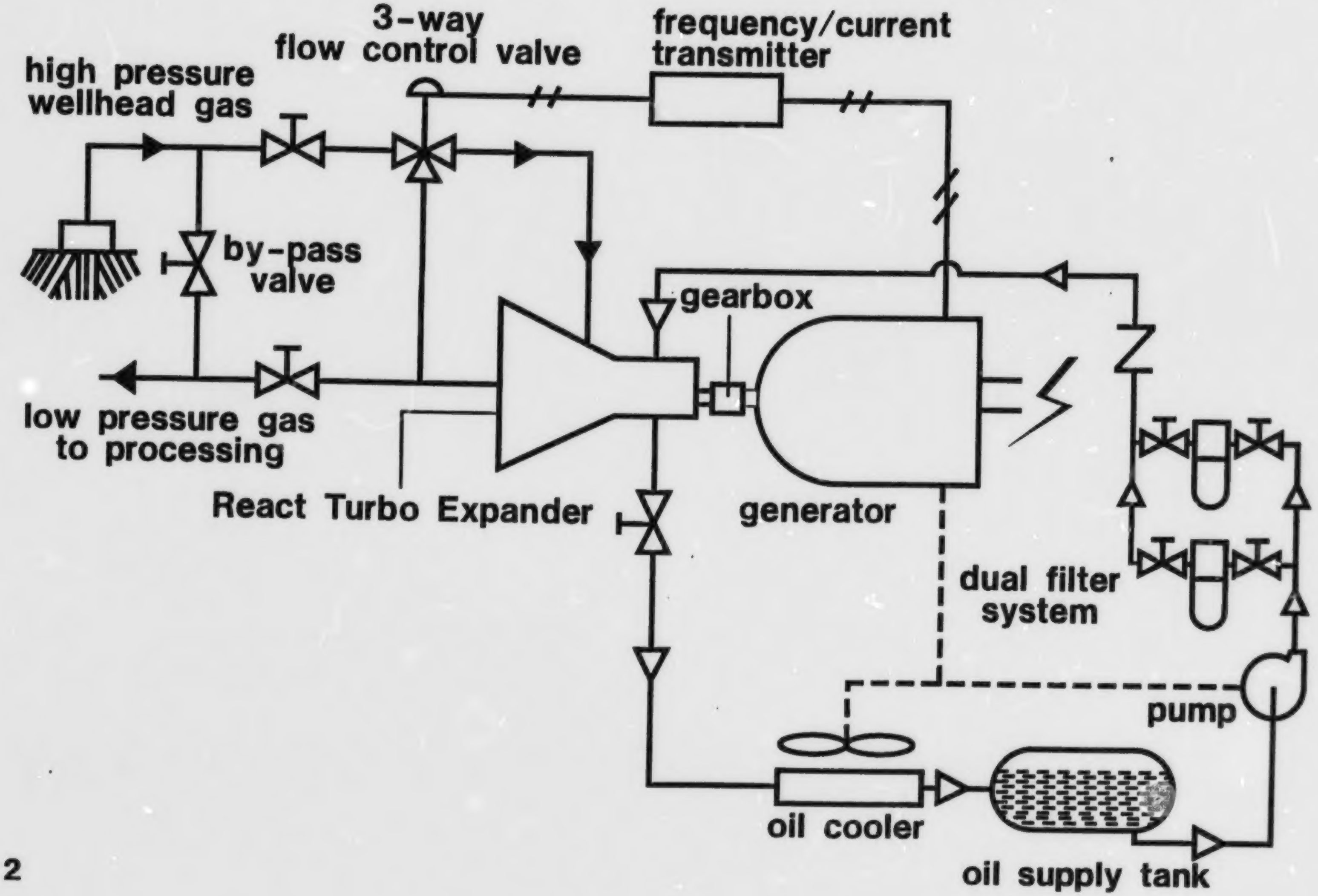

Gas Pressure
Wellhead
Process Plant
Compressor Station
Compressor Station
Compressor Station
City Gate
Station
Distance From Wellhead

Fig. 1

Typical Schematic Wellhead Application



3-way
flow control valve
frequency/current
transmitter
high pressure
wellhead gas
by-pass
valve
gearbox
low pressure gas
to processing
React Turbo Expander
generator
dual filter
system
pump
oil cooler
oil supply tank

Fig. 2

Pressure Reduction System

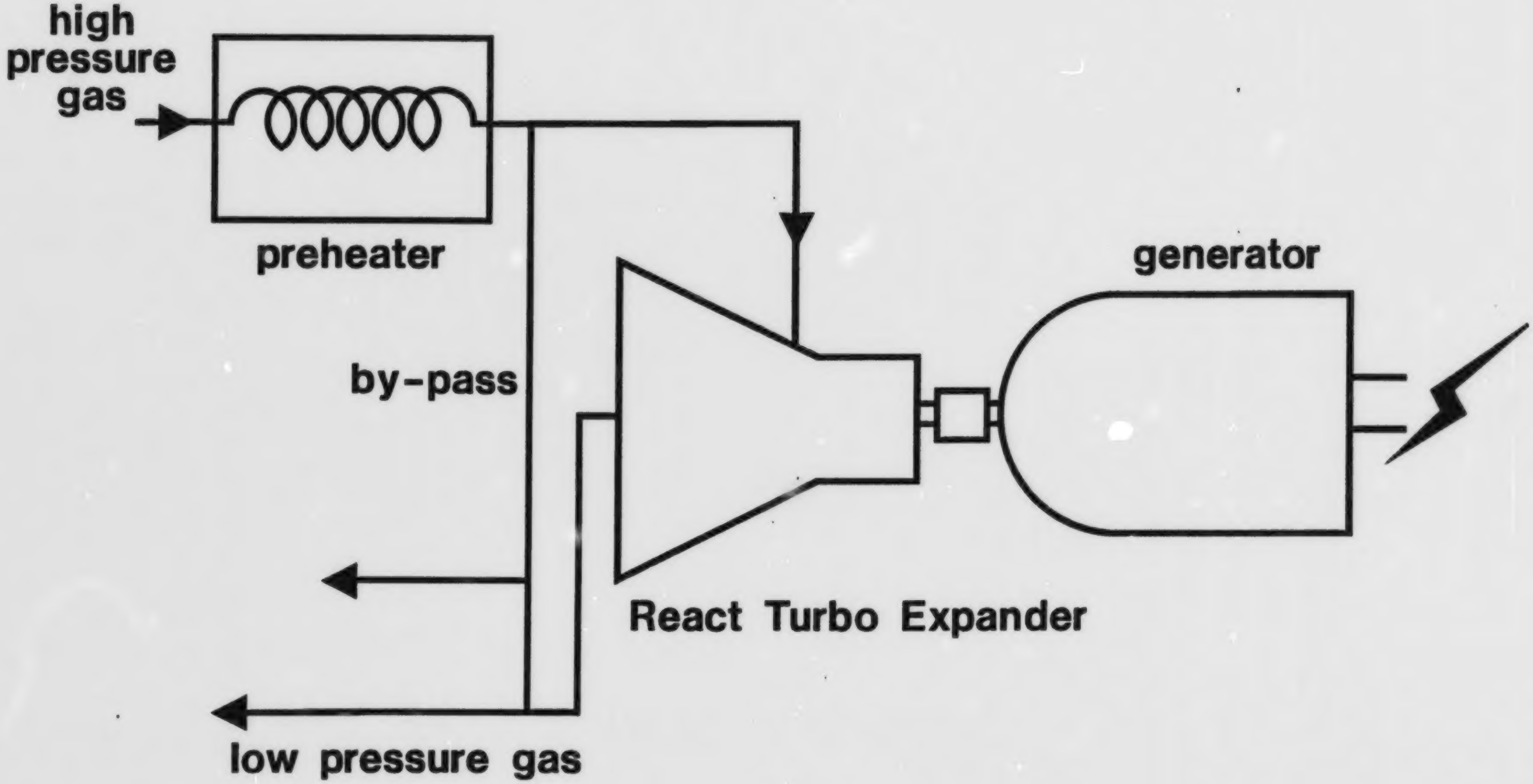

high pressure gas
preheater
by-pass
generator
React Turbo Expander
low pressure gas

Fig. 3

Pressure Reduction Stations

Distribution of Power Output

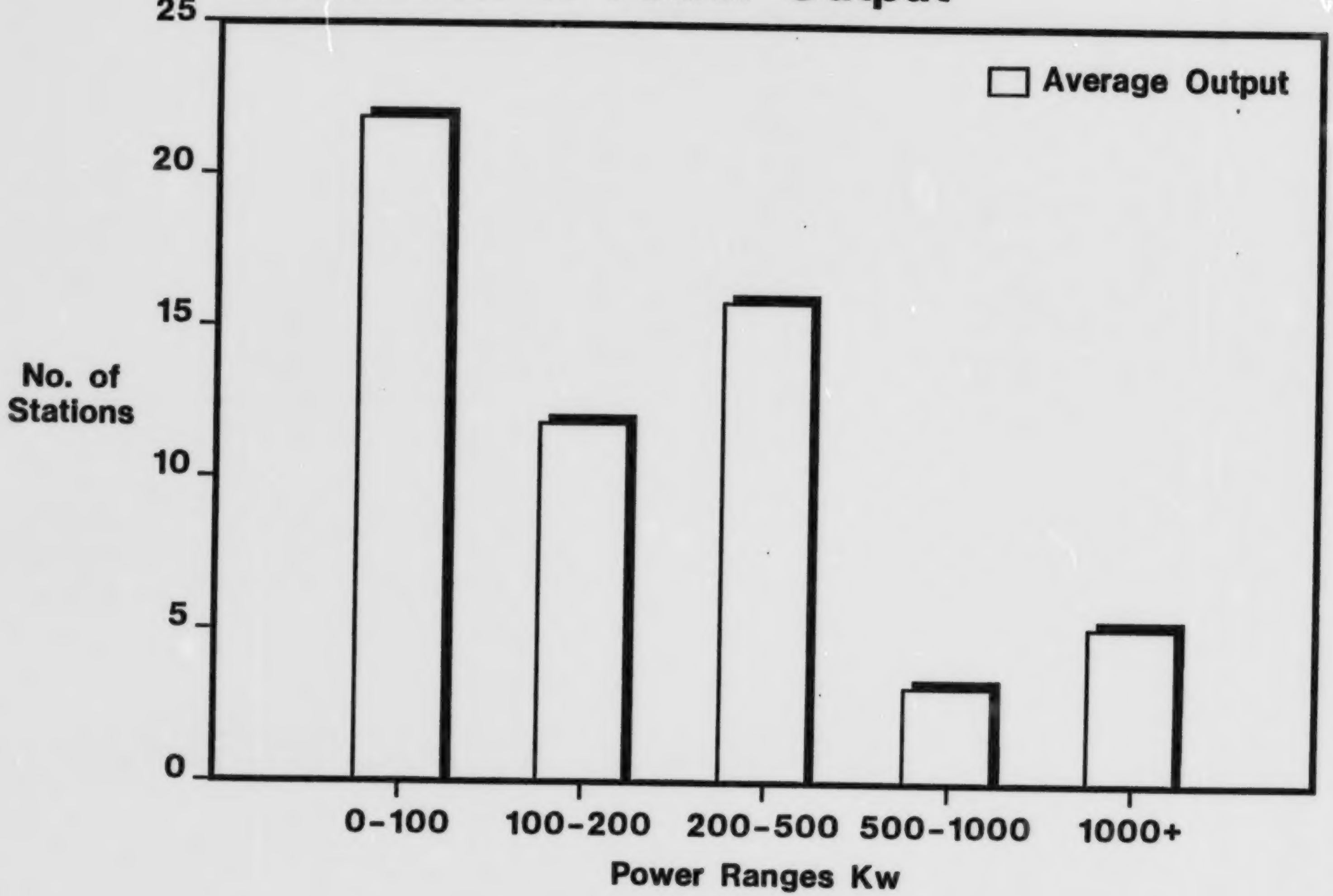
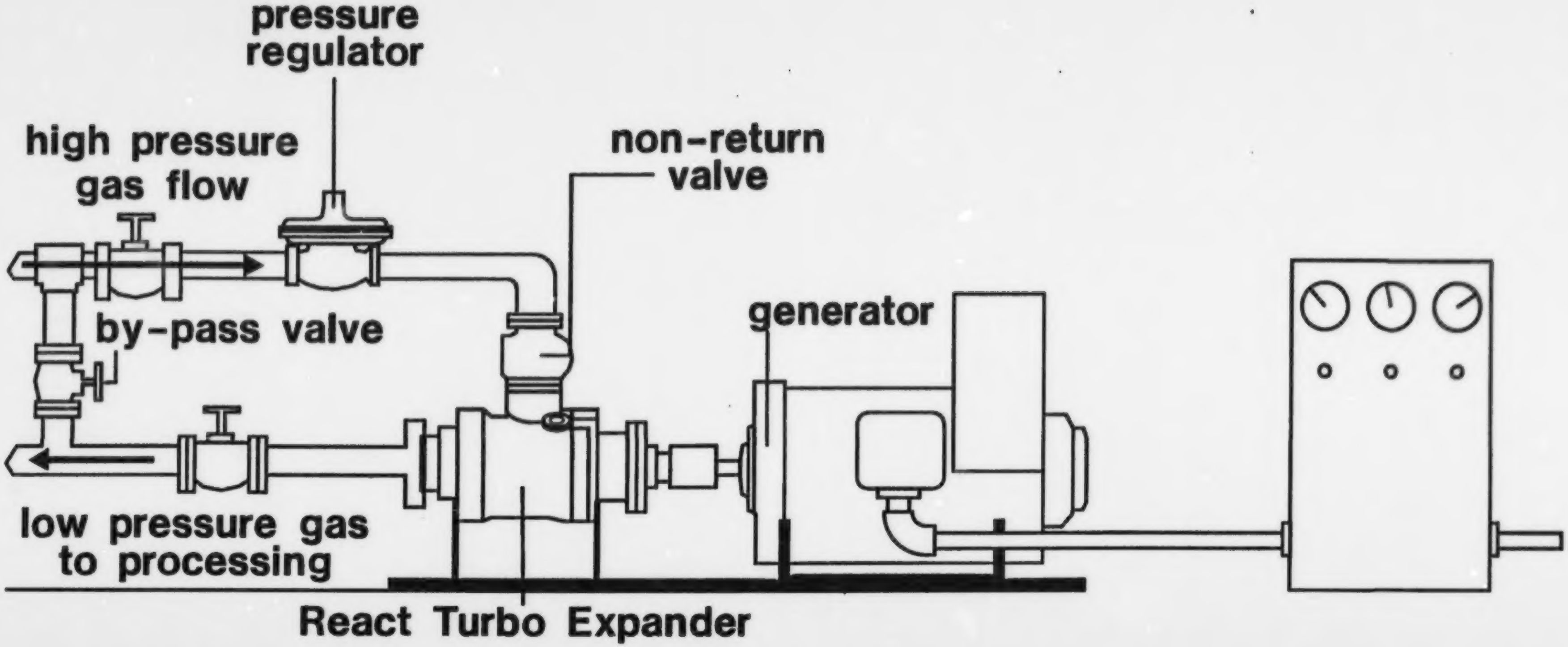


Fig. 4

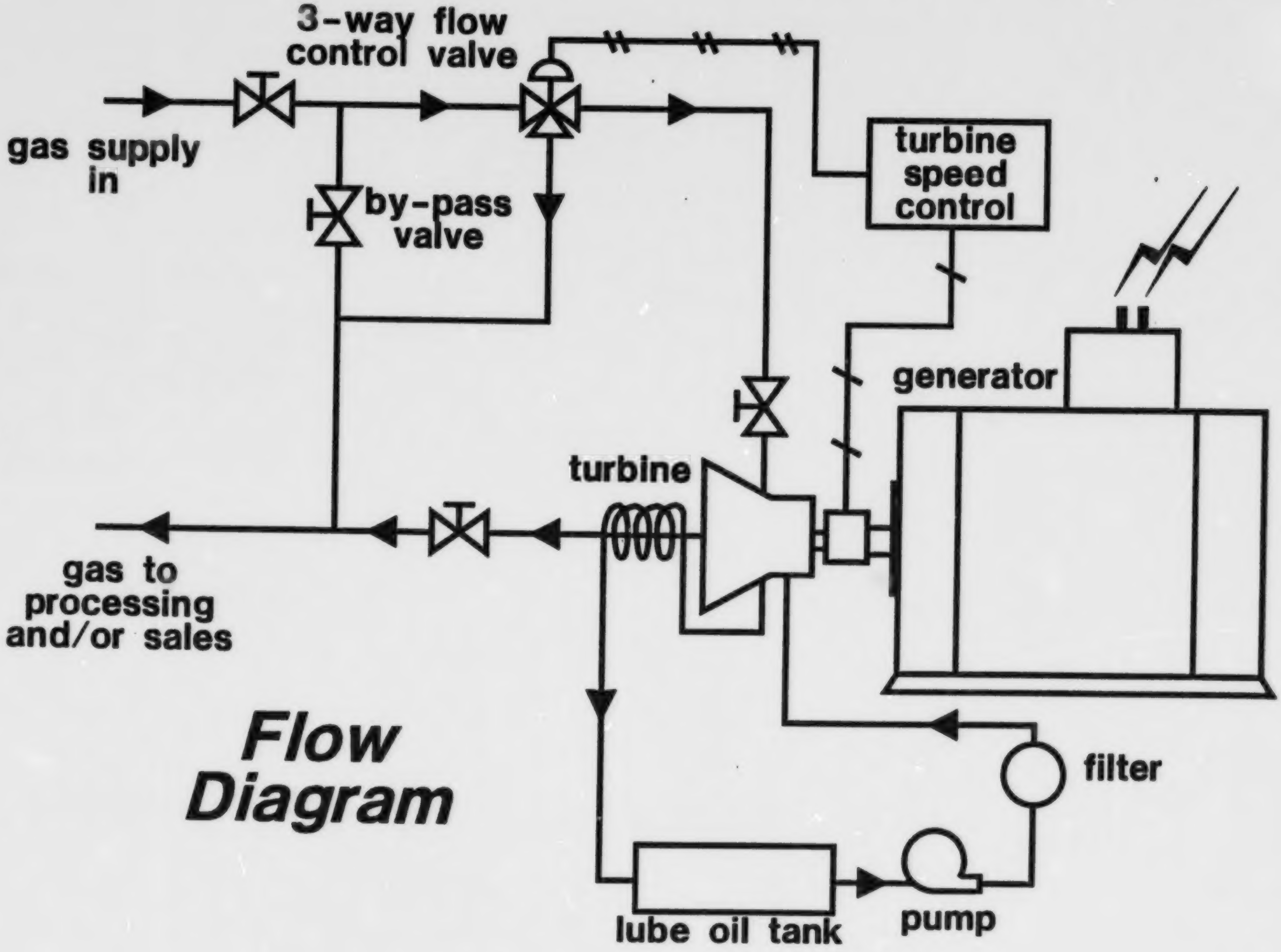
Typical Let-Down Application



pressure regulator
high pressure gas flow
non-return valve
generator
by-pass valve
low pressure gas to processing
React Turbo Expander

Fig. 5

Turbo Expander for Pressure Letdown


3-way flow control valve
gas supply in
by-pass valve
turbine speed control
generator
turbine
gas to processing and/or sales
filter
lube oil tank
pump

Flow Diagram

Fig. 6

Power is Proportional to Flow

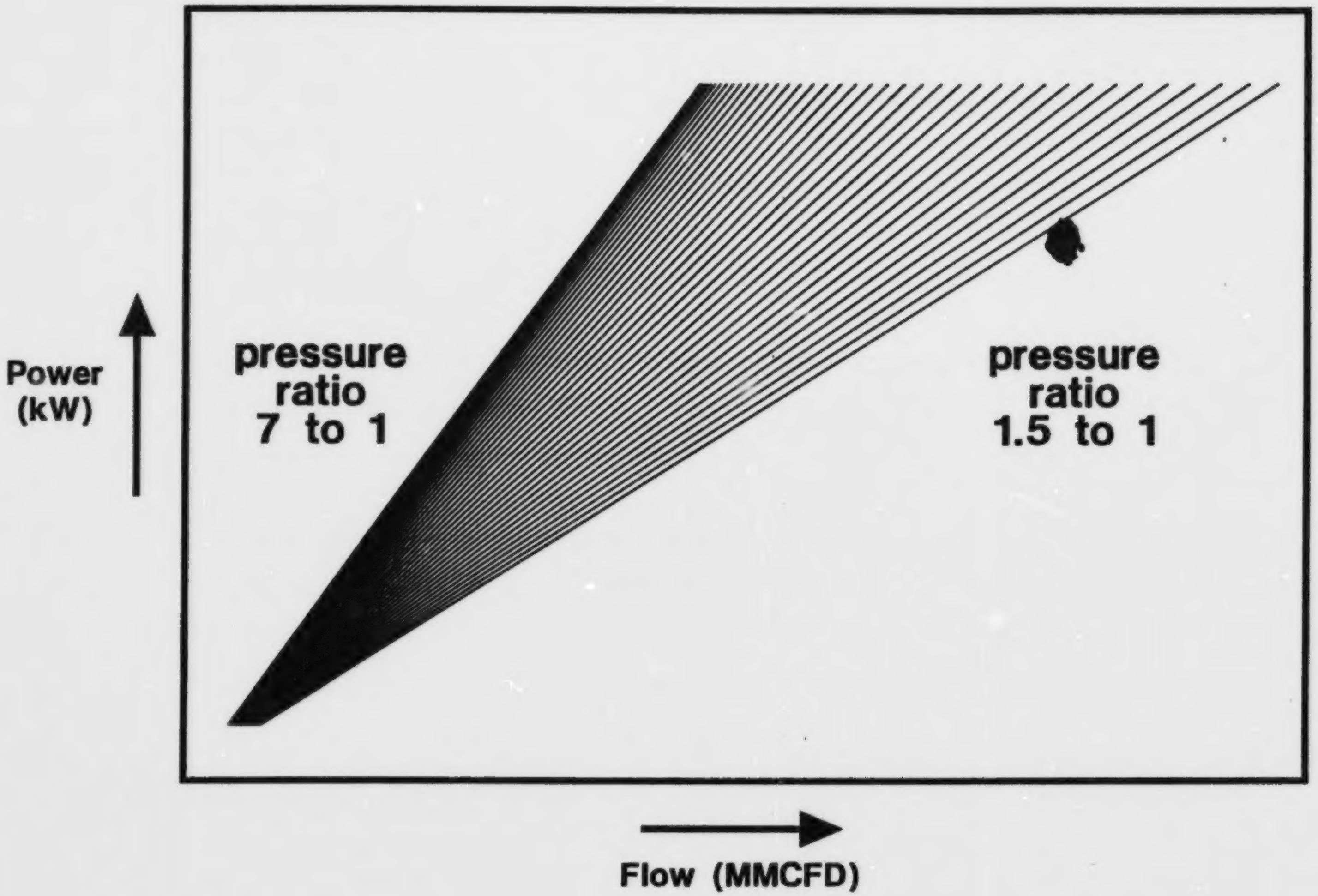


Fig. 7

Turbo Expander - Energy Balance

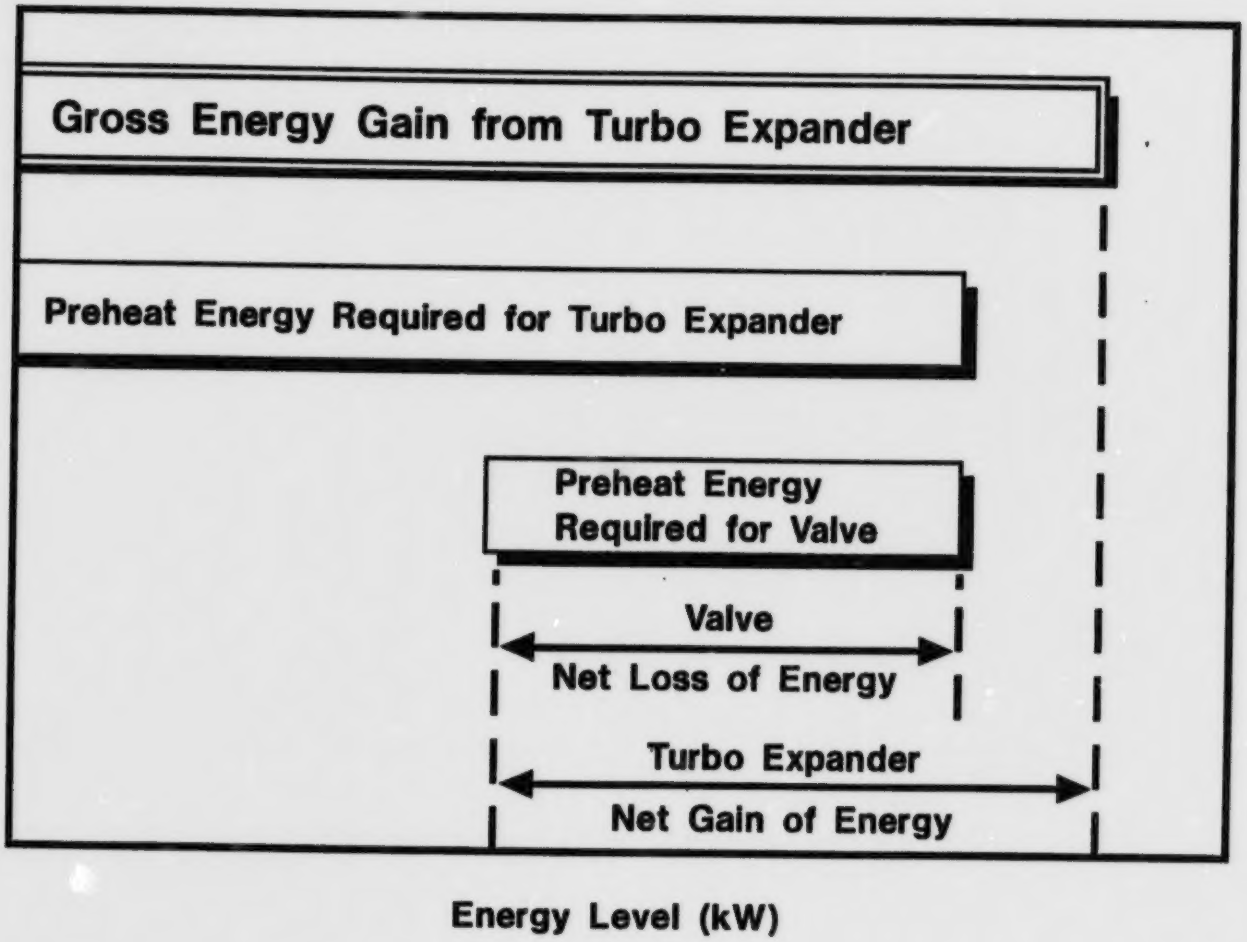


Fig. 8

Daily Flow Profiles
Showing Turbo Expander Maximum Power Limit

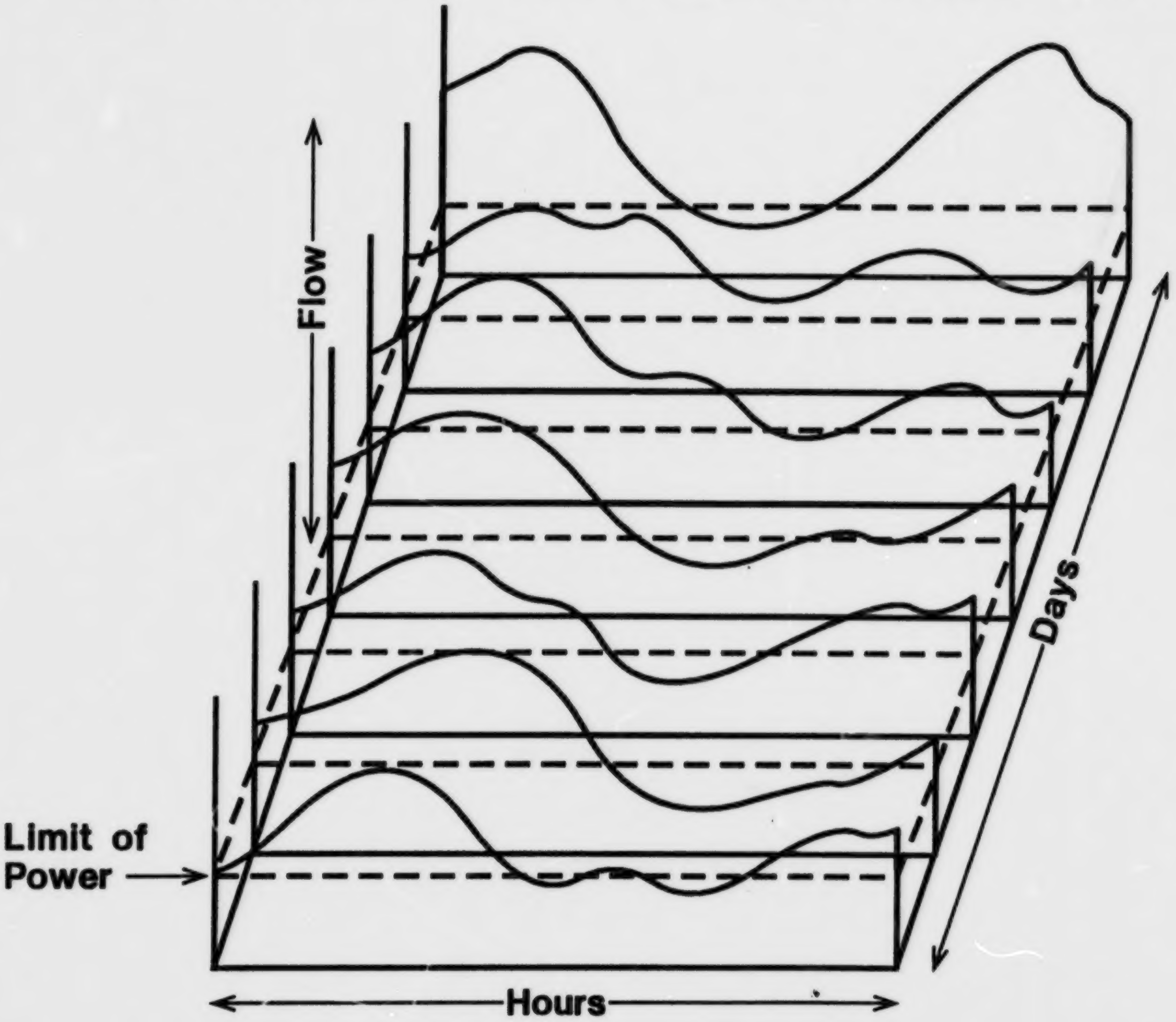

Flow
Days
Limit of Power
Hours

Fig. 9

SECTION XIII

HEAT RECLAMATION FOR HIGH TEMPERATURE FURNACES

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Frank Gilbert
North American Manufacturing Company

9/5/666

ABOUT THE SPEAKER:

Frank C. Gilbert is a 1957 graduate of Alfred University, New York, with a Bachelor of Science Degree in Ceramic Engineering.

On graduation, Frank spent some six years with Union Carbide Corporation, Carbon Products Division, and held the position of Furnace Engineer.

He joined North American Mfg. Co. in 1963 and has since held a number of positions including Combustion Sales Engineer, District Sales Manager and Manufacturing Manager.

In 1981, Mr. Gilbert assumed his present position of Vice President - Sales/Marketing.

North American Mfg. Co. has been involved with the science (and art) of industrial combustion since 1917, and has established a world-wide reputation in this field. The company is a major supplier to industry of combustion and related products such as burners, valves, heat reclamation equipment and electronic control devices and systems.

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HEAT RECLAMATION FOR HIGH TEMPERATURE FURNACES

James D. Bowers, Technical Services

North American Mfg. Co.

Cleveland, Ohio

Efficiency of a fuel fired furnace varies with exhaust temperature and fuel/air ratio. Even with proper burner adjustment, conventional high temperature furnaces have considerable flue losses. As an example 52% of the heat from combustion goes out the flue in an 1800 F furnace--only 48 % is available to heat the work, balance wall and opening losses, etc. Higher operating temperatures mean hotter flue gases and greater flue losses. 2400 F waste gas temperature means 69% heat loss from flue gases. Efficiency can be improved by preheating combustion air with waste gases.

There are two basic types of heat exchangers for waste heat recovery: Recuperators and Regenerators. Recuperators have metallic or ceramic walls that separate two gas streams. Heat from flue gases is transferred through the partition to combustion air. As a result recuperators must be constructed of materials that do not leak and can withstand high temperatures. Usually construction materials are expensive and good for one but not both requirements, making recuperators susceptible to catastrophic failure.

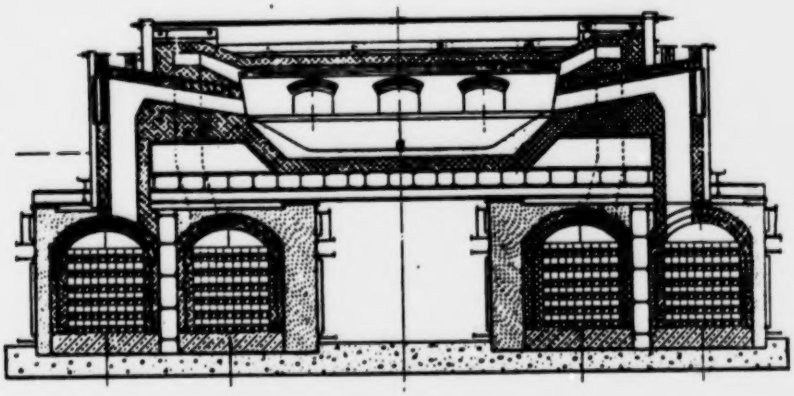

Figure 1: Furnace with traditional regenerator.

Regenerators consist of two chambers, each containing a permeable storage bed of ceramic or metal. Flue gas flows through one chamber giving up heat, while combustion air flows through the other absorbing heat. After a period of time, flows are switched, and air flows through the heated bed while flue gas heats the other chamber in preparation for the next cycle. These flow reversals occur at regular intervals of 15 to 20 minutes.

Regenerators have been used in steel and glass industries since the 1850's. Although highly efficient, their usage has been restricted due to high initial cost and large size. They are often larger than the furnace they serve and require large exhaust fans. Regenerators have far higher heat recovery efficiency than recuperators. There is no worry about leakage since no barrier is needed between combustion air and flue gases.

NEW DIRECT FIRED BURNER

Compact burners have been developed for industrial process furnaces that surmount regenerator size and cost problems. Developed in a joint venture with the Gas Research institute, the burners are described as heat reclaiming burners since they are radically different in design and operation from traditional regenerators. Their bed material has a much larger surface-to-weight ratio. Cycle times have been shortened drastically from 20 minutes to 20 seconds, permitting a radical reduction in physical size. Burners operate in pairs, with one firing while the other acts as flue and heat absorber.

Because of the large surface area in the heat storage bed, combustion air preheats are within 150 F of flue gas temperature. So much heat is extracted from exhaust gases that, they are cool enough to be handled with ordinary steel pipe and rubber flexible couplings on a 2400 F furnace. Exhaust temperatures are lowered to just above the dew point, which not only signifies superior fuel efficiency but also permits use of induced draft fans or venturi eductors of standard low cost materials.

Air preheat temperatures frequently in the 2000 F+ range, produce very high flame temperatures. Because flame radiation varies as the fourth power of the absolute temperature, resultant heat transfer is increased. The very high air temperature causes most gaseous fuels to undergo thermal cracking, which turns the flame luminous and further enhances radiation.

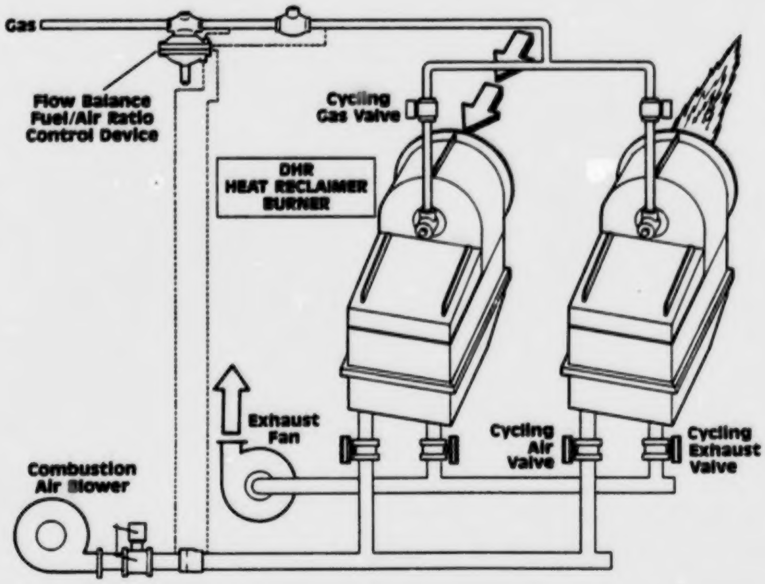

Figure 2: Direct fired burner piping schematic.

The direct fired burner system uses an exhaust fan to draw flue products through the bed of the non-firing burner. Every 20-30 seconds, the cycling air, exhaust, and gas valves reverse. During the second half of the cycle in the illustration, the right hand burner would be firing and the left burner exhausting.

Responding to signals from a furnace temperature controller, control is achieved in one of three ways:

1. On-off--All three cycling valves (air, exhaust, and gas) operate or are off.

2. Modulating--A motorized valve controls combustion air input, and fuel follows via cross-connection to the mass flow ratioing device.

3. Fuel only--That cross-connection is "bled" by a motorized control valve in the impulse line.

In both #2 and #3, the cycling valves continue reversing burner input and exhaust every 20-30 seconds.

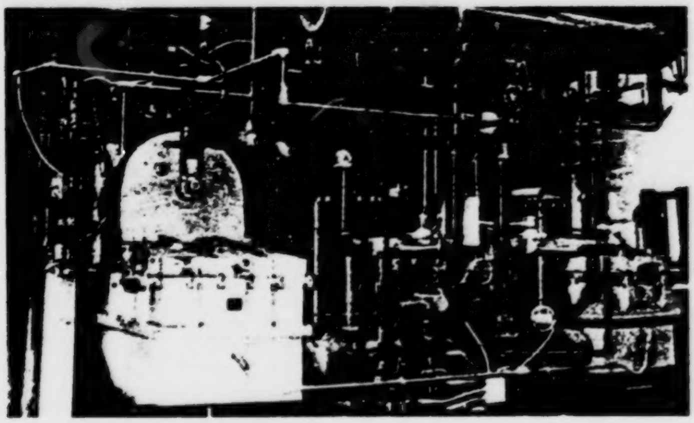

Photo 1: Direct fired burner on aluminum melter.

To illustrate the improvement in efficiency, compare three systems: 1) a conventional burner system with no waste heat recovery (cold air burner); 2) burner system with a ceramic matrix recuperator; and 3) a heat reclamation burner system. The comparison is made on a furnace with 2300 F exhaust gases firing natural gas. Performance is based on equipment operating at 80 % of maximum. Although higher preheats can be obtained at lower flows, dollar return on investment shrinks. Thus 80 % was felt to represent a typical selection for optimizing pay-back.

The cold air system is the least efficient with an available heat (best possible efficiency) of 29 %. The ceramic matrix recuperator improves available heat to 55 % because it preheats combustion air to 1200 F. The heat reclaimer type burner has the highest available heat of 77 % because it heats combustion air to over 2000 F. The ceramic matrix system saves 47 % over the cold air system while the heat reclaimer burner system saves 63 %. Comparing the heat recovery systems, the heat reclaimer burner saves 28 % over the ceramic matrix recuperator.

Savings for a particular furnace should be based on actual equipment sizing, usage, temperature, firing rates, and resultant air preheats delivered to the burner.

FURNACE MODERNIZATION

A comparison of car bottom furnaces at a Pennsylvania forge shop illustrates the application of a heat reclamation system. A load of 164,000 lbs (1 or 2 ingots) must be heated in 72 hours to forging temperature (2150 F).

The previous furnace had conventional burners located near the roof, because there was inadequate space to fire beneath the load. The bottom of the ingots were not heated adequately unless they were removed from furnace, rolled over, and turned end to end.

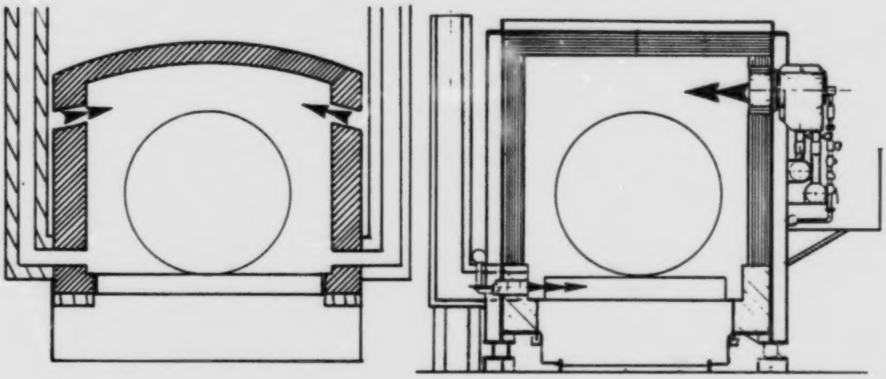

Figure 3: Old furnace on left. New furnace on right.

A new furnace uses six pairs of heat reclaimer burners, all on one side, near the roof. There are eight high velocity burners on the opposite side near the hearth. The heat reclaimer burners provide high efficiency heating capacity, while the high velocity burners contribute stirring energy for uniformity of temperature and even heating. Stirring burners' input is 10% of the total.

Fuel savings are 40 to 45 percent depending on the heating cycle. The cost of modernization was paid back in one year. In addition, more uniform heating improves product quality.

GLASS TANK

A unit melter producing soda glass for bottles and containers measures 36' X 10' and has two zones, without physical partition: a 2500 F melting zone and a 2700 F fining zone. 16 gas premix burners were replaced with seven pair of regenerative burners located on opposite side walls of the tank.

Fuel consumption has been lowered from 26 to under 12 MM BTU/hr while production has risen from 40 to 55 tons/day. The result is a 58% reduction in BTU/ton. The tank was actually run at higher production rates with regenerative burners but the glass feeder mechanism choked.

Melting glass can produce particulate (or other) carry over in the flue gases which tends to plug the heat storage bed. Special burner construction is used which permits easy bed removal and replacement with clean material. Once removed the bed material is cleaned by tumbling in a cement mixer. Beds are cleaned every two to three weeks.

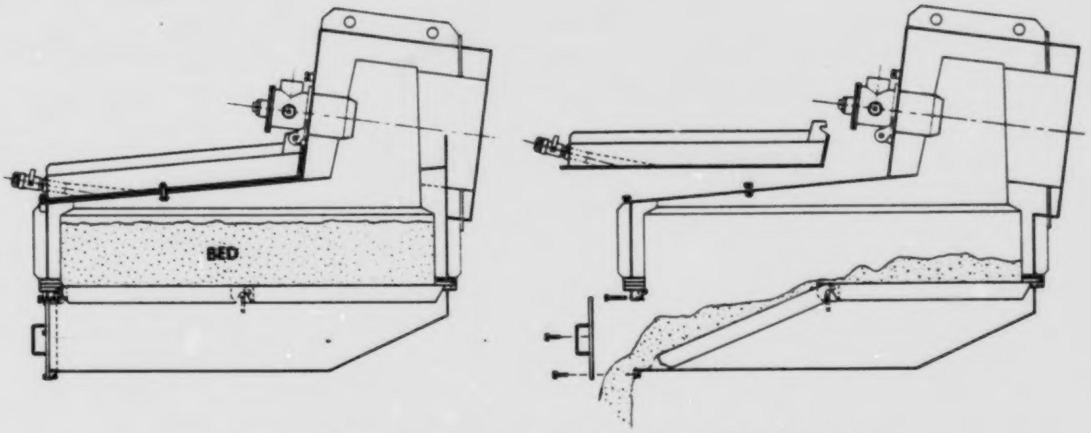

Figure 4: TwinBed burner for quick bed removal and cleaning.

Aluminum reverberatory melting furnaces also can have carry over of particulate. The amount varies with charge type. Furnaces melting clean ingot have almost no carry over while furnaces melting contaminated scrap can have significant amounts. Bed replacement and cleaning can vary from once a week to every other month. Bed material is cleaned by washing with water.

RADIANT TUBE BURNER

Heat reclaiming burners have been developed for radiant tubes, which heat furnaces with protective atmospheres.

The radiant tube burner system uses eductors (rather than an exhaust fan) to maintain a negative pressure in the radiant tube. Air flows to both burners all the time: When the cycling exhaust valve is closed, combustion air is forced through that burner's bed and is preheated to near tube temperature before it is mixed with gas; during the other half of the cycle, the valve opens and the exhausting air flowing through the eductor creates a suction.

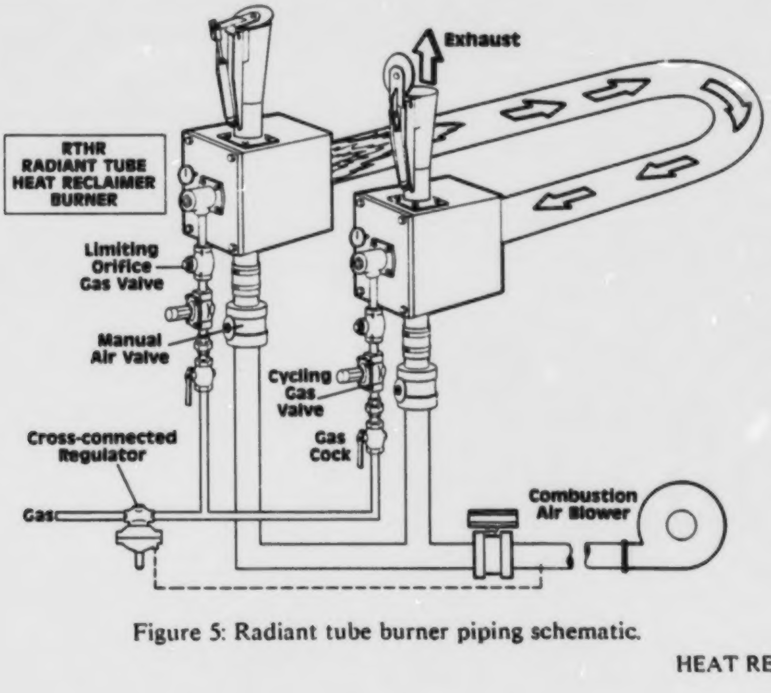
Exhaust
RTHR
RADIANT TUBE
HEAT RECLAIMER
BURNER
Limiting
Orifice
Gas Valve
Manual
Air Valve
Cycling
Gas
Valve
Cross-connected
Regulator
Gas
Cock
Gas
Combustion
Air Blower

Figure 5: Radiant tube burner piping schematic.

Control is realized simply by turning off both cycling gas and main air valves in response to signals from the furnace temperature controller.

APPLICATION TO CONTINUOUS FURNACE

A continuous radiant tube galvanizing furnace is heated with 208 "U" shaped radiant tubes. Furnace temperature is 1800 F with an atmosphere of hydrogen and nitrogen. Conversion to heat reclamation resulted in the following:

1) Fuel savings are 47%.

2) Better tube life and more uniform tube temperatures are obtained with heat reclamation's alternate firing. Conventional burners produced hot spots that tend to cause early tube failures.

3) Lower ambient temperatures around the furnace result from reducing tube exhaust temperatures from 2000 F to 375 F. Extra pay for hot furnace work is eliminated. Better maintenance is expected because of the cooler working environment.

4) With the conventional burner, the tube operates under positive pressure. If a tube breaks, combustion gases leak into the furnace, contaminating the atmosphere. When contamination increases to a certain level, steel surface quality is ruined. The heat reclaiming burner operates with a negative tube pressure; So if a tube leaks, furnace atmosphere is not contaminated.

CONCLUSION

High temperature gas fired furnaces have always suffered from low fuel efficiencies. Heat reclamation burners offer state of the art heat recovery for direct fired and radiant tube high temperature industrial furnaces such as forge furnaces, glass day tanks, metal melting furnaces and ceramic kilns.

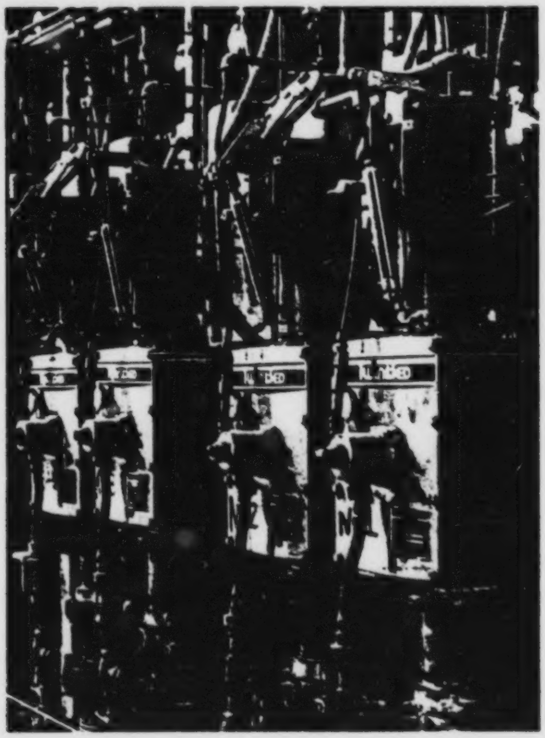

Photo 2: Radiant tube burners on continuous galvanizing furnace.

SECTION

XIV

IMMERSION BURNERS FOR METAL HEATING

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Ron Cockfield
Cockfield Wire Services

9/5/666

ABOUT THE SPEAKER:

Ron Cockfield received his metallurgical training in the steel industry in the United Kingdom, and came to Canada in 1963.

Before forming his own company in North Vancouver, Ron was with the Steel Company of Canada in Hamilton, Ontario, and Davis Walker Corporation of California.

One of Ron's specialities is quality control methods and he has taught the subject at the B. C. Institute of Technology in Burnaby, B. C.

Cockfield Wire Services was formed in 1981 to meet the needs of the metal-working industries in a number of specialty areas such as wire drawing, galvanizing, acid regeneration in metal pickling, wire forming and chemicals used in metal forming and for corrosion protection.

Burns Engineering of Birmingham, England, is one of the companies represented in North America by Cockfield Wire Services. The Burns Company manufactures a recuperative immersion burner for application in galvanizing, zinc and aluminum die casting and melting, salt baths, lead, magnesium, etc.

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IMMERSION BURNERS FOR METAL HEATING

INTRODUCTION

The purpose of the paper is to describe a new design of gas immersion burner used for the melting and heating of non-ferrous metal baths as encountered in the die casting, galvanizing and heat treatment industries.

The burner is known as the Burns, Recuperative Immersion Burner. It was developed jointly by British Midlands Gas Research and B.N.F. Metals Technology an industry sponsored research company. The burner is manufactured under licence by Burns Engineering, U.K.

The burner has been in commercial operation since 1981 and currently is in use in 5 countries including the U.S.A. Installations in Canada are planned for 1986. All applications to date have confirmed the initial projections of substantial fuel savings plus other spin-off benefits which we will discuss.

COMPARISON WITH OTHER HEATING METHODS

Traditional methods of heating in the industries mentioned typically employ open flame burners, heating indirectly through a metal container or through the molten metal surface.

Examples in the die casting industry would be the crucible and reverbatory type of furnace. Galvanizing baths are heated by flat flame burners, high velocity recirculating burners and in the case of ceramic baths, top fired designs are employed.

The economics of these heating methods and their impact on the furnace design are well known. Typically these methods suffer from some or all of the following -

- low energy efficiencies, less than 30% heat to metal
- regular replacement of the containment vessel
- contamination of the melt with oxides
- cumbersome and expensive recuperation systems and for some furnaces not practical

The new burner design offers solutions to all the above problems.

DESCRIPTION OF BURNER AND OPERATION

The burner is comprised of two major elements. (Fig. 1)

- a recuperative burner head
- a closed, nitride bonded silicon carbide tube.

Air and gas are introduced separately and mixed at the nozzle. The burner fires into the inner ceramic tube and circulation of the combustion gases takes place between the inner and outer ceramic tubes. The outer tube has high thermal conductivity and transmits heat very efficiently and uniformly to the molten metal surrounding the tube. The convection forces introduced are very strong and provide extremely uniform bath temperatures.

Recuperation is achieved by exiting the exhaust gases through a carefully designed shell heat exchanger integrated in the burner head so as to pre-heat the combustion air. Flue gas temperatures are thus reduced and the need for cumbersome external recuperators is eliminated. Each burner can be isolated from any other and the burner controls provide fully modulated control (no high fire-low fire), full flame supervision and automatic start up, for each individual burner. Gas/air ratios are closely controlled over the whole range by use of a special 3 diaphragm regulator responding to both gas and air flow rates.

BURNER CAPACITY AND EFFICIENCY (Fig. 2)

The graph shows the gross inputs and net heat release at various temperatures. For typical zinc applications (460°C) the burner is rated at 2½ therms gross input with a heat release of 1.4 therms per hour, an efficiency of over 60%.

APPLICATIONS (Fig. 3)

The applications to date cover single burner installations for melting and holding furnaces in the die-casting industry.

3-6 burner furnaces for zinc melting and zinc dust production.
6-14 burner installations for continuous and batch galvanizing.

In addition we are now seeing the first installations of these burners fitted with metal tubes for salt bath heating and very shortly for lead baths.

ADVANTAGES

Compared to the indirect methods of heating, the immersion burners have provided the following benefits -

a) 25-40% energy savings
b) reduced contamination of the melt from oxides
c) improved temperature control and uniformity
d) low refractory wear above the melt due to lower air temperature.

INFLUENCE ON FURNACE DESIGN

Apart from the direct benefits, the introduction of these burners has led to changes in furnace design which has resulted in further operational savings.

For example, in the galvanizing industry traditionally 2 types of furnaces have been available.

- steel kettle, side fired with flat flame or recirculating burners.
- ceramic kettle - top fired with burners above the zinc.

STEEL KETTLES

These are prone to failure due to zinc attack accelerated by high wall temperatures particularly when working at capacity. Bath lives of 1-2 years are typical in continuous lines. The immersion burner eliminates high wall temperatures or hot spots and allows the outside to be completely insulated. We would expect bath life to increase dramatically.

CERAMIC KETTLES

Bath life is not a problem, however, the top-fired design results in very large bath volumes, high energy usage and heavy zinc ash formation. Immersion burners reduce the bath volume by up to 40%, energy costs by up to 40% and eliminates the zinc ash formation.

In fact we believe for many new galvanizing installations the steel kettle is made obsolete by the combination of a ceramic bath and immersion burners.

CASE HISTORIES

Die Casting - U.K.

A single burner aluminum holding furnace specially designed to suit the immersion burner placed alongside a traditional crucible furnace provided fuel costs of only 1/8th and paid for itself in 7 mos.

ZINC MELTING FOR ZINC POWDER AND DUST - U.K.

A two burner furnace working 24 hours a day gave fuel savings of 48% compared to a crucible-type furnace feeding an adjacent plant.

WIRE GALVANIZING - U.S.A.

A top-fired ceramic bath was converted to immersion burners. Output was increased by 40% without increased fuel consumption. The costly problem of zinc ash production has been eliminated.

SUMMARY

The Burns Recuperative Immersion burner is an alternative to the traditional indirect heating methods employed in the heating and melting of non-ferrous metal baths. Offering direct savings in energy costs of up to 40% and potential benefits in improved furnace design and maintenance costs.

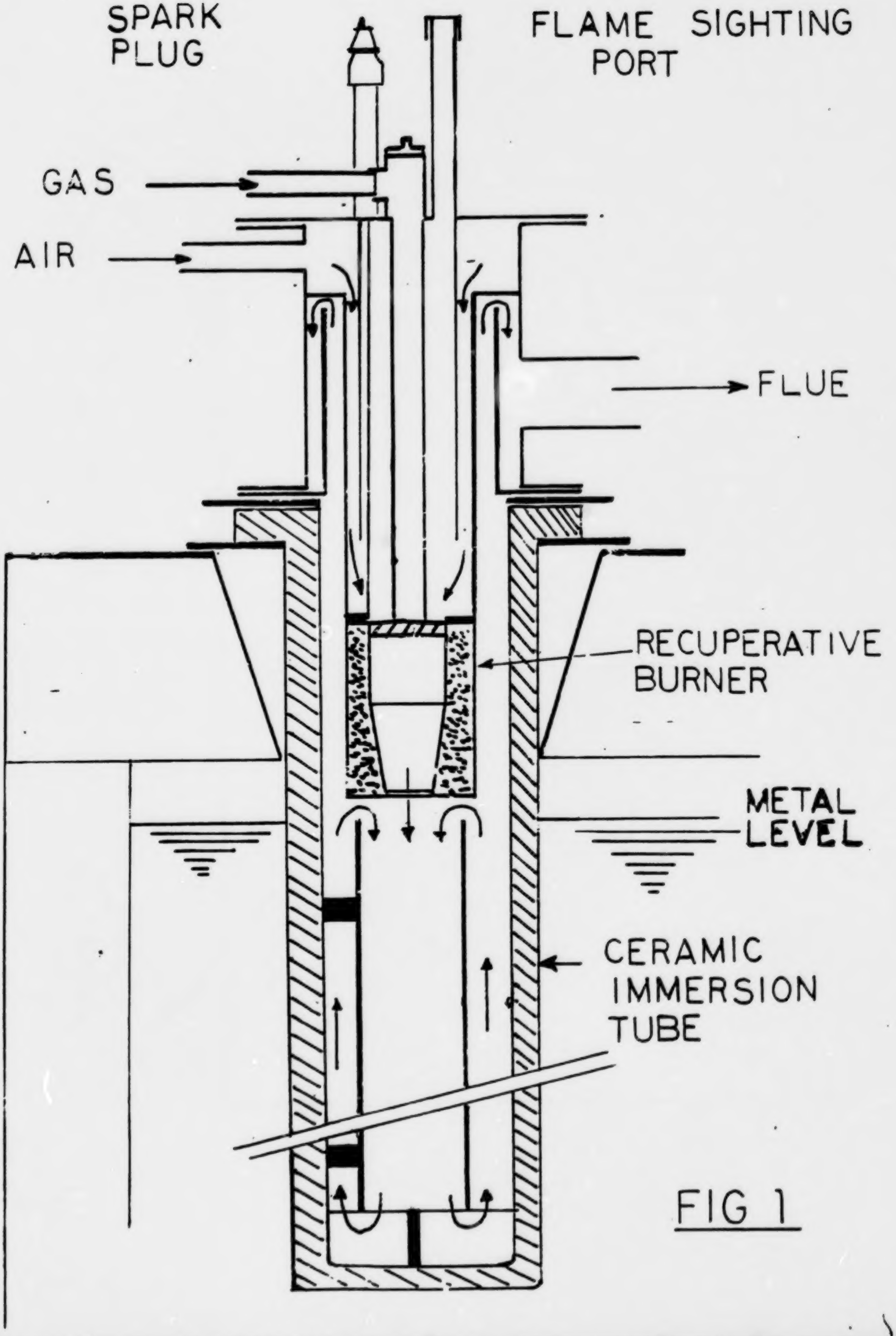
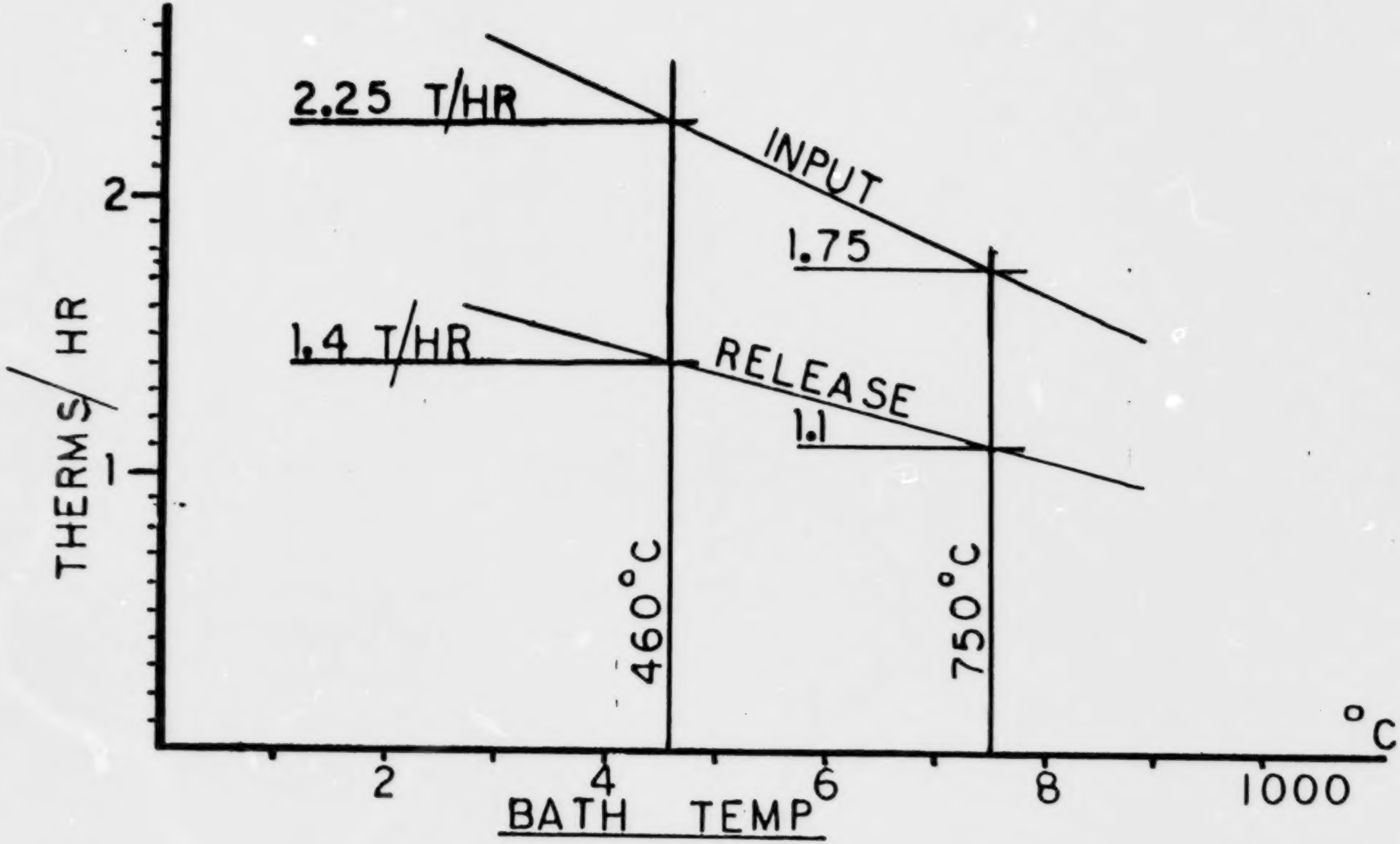

SPARK PLUG
FLAME SIGHTING PORT
GAS
AIR
FLUE
RECUPERATIVE BURNER
METAL LEVEL
CERAMIC IMMERSION TUBE

FIG 1



RATING/HEAT RELEASE OF 2¼ T BURNER

FIG [illegible]

APPLICATIONS

1/ DIE CASTING

- ZINC
- ALUMINUM
- MAGNESIUM

2/ MELTING FURNACES

- ZINC ALLOYING & REFINING
- ZINC DUST & POWDERS

3/ GALVANIZING

- CONTINUOUS WIRE & STRIP
- BATCH
- HIGH TEMPERATURE

4/ HEAT TREATMENT

SALT BATHS
LEAD BATHS

FIG 3

SECTION

XV

HI TECH BURNER DEVELOPMENTS

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B.C.
13 - 14 April, 1988

by Broc Probert
Bloom Engineering Company Inc.

ABOUT THE SPEAKER

Broc Probert received his engineering training at Marion College, San Francisco.

He is Westcoast District Manager for Bloom Engineering and is responsible for sales and engineering for Western U.S.A. and Canada.

Broc worked for Bethlehem Steel and Kaiser Steel as Power House Engineer, Combustion Engineer, Maintenance Engineer and Utilities Foreman for more than 25 years.

Bloom Engineering Company was founded 57 years ago and is involved in the application, design and manufacture of specialized combustion and related equipment for the basic ferrous and non-ferrous industries, metal fabricators, the glass industry, the refractories industries and many others.

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9/5/652

Hi Tech Burner Developments

High tech burner development in recent years has been a combination of innovative design and, even more importantly, refinement of existing design components in order to integrate these improvements into precise, effective and trouble free heating systems. An innovative design component whether it be a burner, a control device, a new furnace refractory or furnace accessory such as a greater flow recirculating fan or aluminum pump will often precipitate refinement of complementary devices in order to take full advantage of the new capabilities.

Principal areas of advancement in burner design have been:

A) Low pollution level burners. Low NO_x/CO;

B) High turndown burners;

C) High efficiency burner designs;

D) High heat release designs;

E) Controlled heat release pattern burners;

F) Direct firing burners to replace indirect firing.

Low Pollution Level Burners

While others have described low NO_x burners in detail at this conference, this aspect of burner design does have and will continue to have a revolutionary effect upon burner design so it will be briefly discussed here as well. While Southern California continues to be at the forefront of NO_x emission reduction requirements in the U. S., it is certain that more stringent requirements will be legislated in other states and provinces. When one looks at the common means of controlling NO_x, one sees that the reduced flame temperatures and/or staged combustion methods would most probably have an adverse effect if typical existing burner models were used. It is obvious that low NO_x burners must be purpose designed. The goal of responsible equipment manufacturers has been not to allow heating performance to suffer but to design a low NO_x burner that compares favorably in other respects to state of the art conventional burners.

One version of a low NO_x burner is shown as exhibit 1. This burner utilizes a recirculation area about the burner air distribution baffle to induce products of combustion into the air/gas mixing area, thus reducing flame temperature and resultant thermal NO_x generation. Purpose designed low NO_x burners do not necessarily have to degrade burner performance. Our type F low NO_x radiant tube burner exhibits a superior tube temperature characteristic which will enhance radiant tube life as well as reducing NO_x to extremely low values. This is a second generation low NO_x radiant tube burner which halves again the already halved NO_x levels of first generation low NO_x radiant tube burners.

Our company is fortunate in having a Japanese affiliate company and a major European subsidiary company where NO_x regulations have been in force longer and have been more severe than in North America and where considerable experience has been gained with low NO_x burner design.

CO generation, aldehydes and other pollutants as well as NO_x, will be subject to control and, as in the case of NO_x, may require the replacement of existing burners to meet government imposed standard.

High Turndown Burners

With the adoption of furnace pressure control, fuel and air metering systems that can accurately measure small volumes, furnace supervisory systems and the use of low heat loss fiber insulation, the subsequent need for high turndown burners has evolved. The term high turndown, as used here, refers to burners that can turn down from 10 or 20 to 1 without resorting to large quantities of excess air. Bloom has developed a number of burners that have this capability. Certain of these burners exhibit other desirable characteristics as well, such as maintained product velocities, flame shape control and/or luminosity control. The first burner shown (exhibit 2) is of our maintained energy design. It is, in essence, a burner within a burner. There are two concentric burner air baffles. When firing the burner at capacity, the inner baffle air supply line is valved off, and all air is applied to the outer baffle. As burner flow decreases below 25% capacity, the valve to the inner baffle is opened and the outer baffle valve is closed. (Exhibit 3.) The use of this burner assures that the burner flame is maintained in direction and forward momentum of burner products without the typical flame tip lifting of a conventional burner when turned down. (Exhibit 4.) This burner may be turned down typically 16/1 with 5-10 percent excess air. This burner type maintains proper fuel/air mixing by utilizing the energy in the combustion air. This is accomplished by changing the size of the burner orifices and burner part when going from high to low fire and vice versa.

Another means of maintaining the fuel/air mixing ability of a burner at high turndown is by using the potential energy of high pressure fuel gas. While gas is only six-tenths as dense as ambient combustion air and is only present in amounts approximately 10 percent of the air, natural gas pressure is generally available in the 2-5 PSIG range, which is considerably in excess of normal combustion air pressures. Bloom has developed a series of Fuel Energy burners, some for specific applications. The simplest version uses a high turndown lance (exhibit 5). A typical burner of this design would operate with a small fraction of the gas through the lance (high pressure) and the balance (low pressure) through the main gas connection. The high-pressure lance is left on over the operating range of the burner once the burner has been lit. The lance gas supply is taken off after metering but before the fuel control valve so a volumetric system can account for the fuel flow. When using pressure balance control, a bias is set into the regulator to accommodate lance gas. Turndowns as high as 20/1 can be achieved with this burner. Often turndown lances can be retrofit to existing Bloom burners.

Heat treat furnaces exhibit a need for high turndown as well but also require a maintained circulation of furnace gasses over the turndown range. Years ago, the tempered flame method of firing was developed for use with critical uniformity applications. This system used a burner, stable under high percentages of excess air. The burner retained its capacity combustion air flow while the gas turned down to meet the temperature requirement. This was really a revolutionary approach as it recognized the need to maintain momentum of burner gasses, during periods of burner turndown, to keep the furnace gasses stirred. Momentum equals mass times velocity. By keeping the burner product flow rate high at turndown (the typical soak condition where uniformity is critical) the momentum was high enough to keep the furnace gasses moving. In those days when 5 percent per therm gas was available, this method was the answer to the heat treater's dream. With gas at many times that cost now, refined methods of maintaining momentum are required as it is now expensive to heat all that air. The impulse or pulse firing system has been used in Europe for some years to improve momentum under turndown conditions. This method uses a control technique which turns burners on and off as heat is required. A burner fires at either 100% or 0% with the air and gas at near ratio conditions at capacity. In a multiple burner system of say 10 burners, at 10/1 turndown, one burner would be firing at a time. The firing burner would be sequenced, i.e., one burner would fire for a short period, then another in a repetitive sequence cycle through all the burners. Since P.O.C. velocity varies as the burner flow rate, the entrainment of furnace gas is greater when 1/10 the furners fire at 100% flow than if 10 burners were fired in ratio and were turned down 10 to 1.

The Uniheat® burner system was designed as a next generation burner to overcome the inherent difficulties with the Tempered Flame Design (fuel cost) and the Impulse Fired System (fce pressure control and equipment cost), as well as providing superior circulation of gasses at high turndowns when one is in the soak portion of the heating cycle where uniformity is particularly critical. The Uniheat®

system utilizes a high velocity burner and an adjacent small air port. (Exhibit 6.) The burner is typically operated in ratio throughout its turndown range. At approximately 25% capacity, the small jet is opened which supplies approximately 25% of the maxiumum air flow through a port which is 1/16 the area of the burner port. We are trading velocity (1/16 the size port) for air volume (1/4 the volume) to give the peak momentum level for the 4/1 turndown and greater ranges. Exhibit 7 shows momentum level versus percent capacity input.

More efficient burners

While we have enjoyed a fuel price respite in recent years, prices in the long term are bound to rise. Our first surge in fuel prices of a few years ago, precipitated development of new recuperator technology which, in turn, spurred development of higher combustion air temperature burners. With air preheat fuel is saved as the combustion air which is heated to flame temperature during the combustion process, doesn't take as much heat out of the furnace when it is partially heated before it enters. Air preheat only saves fuel in that it doesn't require as much for heating as when cold air is used. Air preheats as high as 2000°F have been achieved though the typical recuperator/burner system is generally rated at 1200°F or less. This air preheat can save as much as 30% of the fuel with 1800° waste gasses into the recuperator.

Another efficiency refinement has been the reintroduction of regenerative burners. Modern versions of this earliest means of heat recovery are being applied to many furnace types. Bloom is currently conducting tests in our laboratory on a prototype state of the art regenerative burner and we anticipate developing a version of this design which will combine the fuel saving qualities of this burner type with environmentally acceptable low NO_x characteristics.

Improved Heat Transfer Burners

Improved heat transfer can result in improved burner fuel efficiency. It can also result in faster production or increased production efficiency. A perfect example is a burner that would be applied to a modern high speed aluminum melting operation. Purpose built burners for this application generally have a high (optimum) burner port density. Port density refers to the number of BTUs at capacity per square inch of port diameter. Bloom burners used for this service mix rapidly and provide a flame shape which is both intense and also of a suitably large diameter. This provides a large frontal area, high velocity, high temperature, combusted, burner products stream to envelop and rapidly melt the heavy scrap which would be charged in front of the burner.

Aluminum melting burners in the past have typically had a "clear" flame. The loss of flame luminosity had been considered a necessary trade-off for the superior convective action of the hard driven products. The 1209 series aluminum melt burner (Exhibit 8) allows us to reintroduce the visibly radiant flame component throughout the burner turndown range. This aluminum melt burner has three gas passages:

1) High turndown gas lance - on while burner lit;
2) Inner annulus - more gas, greater luminosity, longer flame;
3) Outer annulus - more gas, less luminosity, shorter flame.

External valving of the gas to the two annuli can be used to vary flame length and/or luminosity over the turndown range, or fixed orifices can be installed in the gas supply lines to instill certain characteristics. This is a state of the art aluminum melting burner. In one installation where the existing burners were modified by adding luminosity control, a 5% fuel savings and a 5% production increase resulted.

Both the burner within a burner and the fuel energy design burners may be used when flame shape control is desired. In a side fired steel mill reheat furnace, for example, it may be desirable to maintain flame length even when the burners turn down. In a longitudinally fired continuous reheat furnace, it may be advisable to shorten flames in the heating zones at certain production rates and lengthen the flame during other conditions. In a rotary kiln, flame length change can be used to accommodate different product flow rates.

The type F low NO_x radiant tube burner improves heat transfer in a radiant tube by releasing heat more gradually cutting off the tube failure causing high temperature peak as shown in Exhibit 9. This is the direct result of exhaust gas recirculation and an example of how low NO_x can mean improved burner performance.

Controlling Heat Release Pattern

All combustion engineers used to refer to flame shape and burner manufacturers would pride themselves on fitting the flame shape to the furnace. It was thought that all flames should be visibly luminous and that these flames should extend over the work. Modern hard driven flames are often "clear", emitting little visible radiation when viewed in a hot furnace. The term flame shape, in this instance, becomes inappropriate and we now have to think of heated products of combustion penetrating the furnace and emitting radiation which remains the preponderent heat transfer mechanism at high temperature. It has been found that the clear hard driven flames exhibit better product heating characteristics than the old lazy longer flames as well as minimizing the impingement on the work of partially burned gasses that contain high percentages of oxygen as well as combustibles. This type of burner tends to stir the furnace atmosphere and aids the convective component of hear transfer. In certain instances, we are able to reintroduce flame visible luminosity while retaining the other benefits of the harder driven flames, which, as previously described in the aluminum melting application, improves radiant transfer still further.

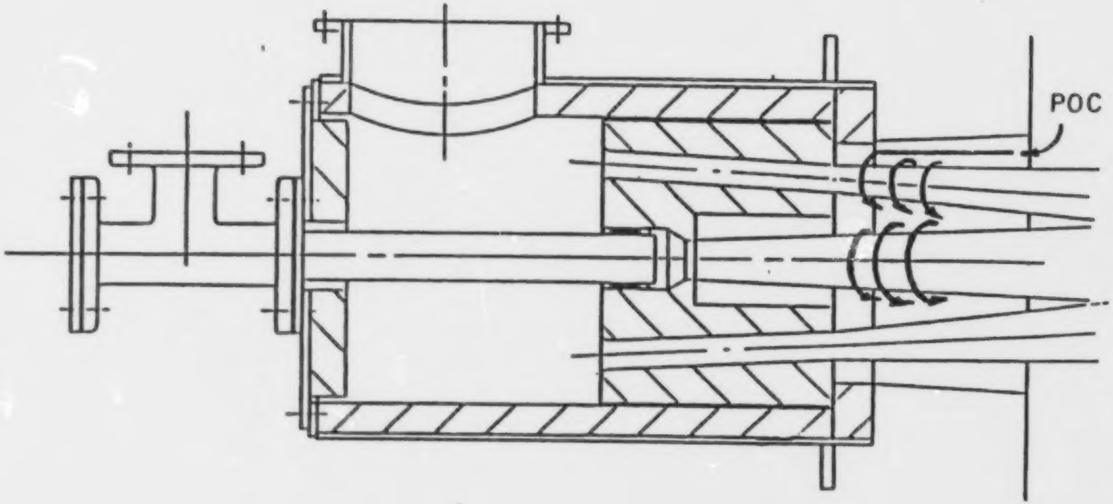
In many instances, contact between the "flame" and the work is undesirable. The problem may result from the extreme temperature of the flame which could overheat areas or surfaces, and/or might be due to the adverse chemical effect of impinging the work with air, fuel and intermediate combustion products. Another undesirable result of flame contact can be the "quenching" of the flame by the work, leaving unburned products when combination of furnaces below the autoignition temperature, stratification of furnace gasses and lack of combustion space might contribute further to this condition. Undesirable flame/ work contact often occurs during turndown when the falloff of fluid mixing energy causes the flame tips to lift and the flame to become lazy. With modern burners such as the maintained energy or the fuel energy burner, flame direction can be maintained under turndown conditions and impingement need not occur. In cases where impingement is desirable such as in aluminum melting operations, an intense fast burnout flame with considerable forward thrust is preferable to impingement of the work by partially combusted gasses.

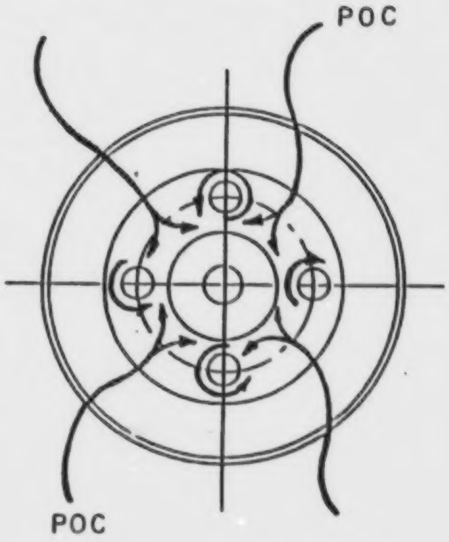
Direct firing rather than indirect

With improvement in burner controlability, many furnaces that at one time used indirect firing, can now be converted to direct heating. Steel mill box annealing furnaces were at one time fired by using radiant tubes to transfer heat to the inner coil covers which contain a hydrogen/nitrogen atmosphere and the steel coils being heated. Most of these furnaces have now been converted to direct firing using HTR type burners which have a disc shaped flame which stays close to the wall heating the burner block which, in turn, radiates heat to the work. This burner (Exhibit 10) provides an excellent radiant heat transfer characteristic which is extremely effective as the radiant component is predominant in a typical furnace environment at high temperatures and emissivities (Exhibit 11). This style burner can be fired with preheated or ambient temperature combustion air. The burner may be used to direct fire galvanize or lead pots. The disc flame characteristic allows the conventional bridge firing walls to be removed and the pot can receive radiation direct from the burner. This burner design is used to fire furnace roofs which then become what one may consider a panel radiation source where the whole panel responds to temperature requirement and provides a well distributed radiation to the work.

Another example of replacement of radiant tube heating by direct heating is in the aluminum homogenizing furnace. Burners have been developed which fire into an open ended alloy can that give the benefit of improved heat transfer and efficiency of direct firing. A burner shown in Exhibit 12 is of all metal and fiber insulation construction and uses no refractory products in the burner assembly so that those gritty materials cannot damage the product being heat treated.

In order to gain utmost effectiveness from a new or retrofit furnace, the project must be viewed from a systems standpoint. While improvements can be made in steps, we must be careful to consider the implications of the new component whether it be burner control, insulation, or other upon the system. It does little good to apply a sophisticated air to fuel ratio control for example, to a furnace where the burners simply cannot mechanically mix the fuel and air properly for complete combustion at higher turndown. New fiber insulation of a furnace may well require additional turndown capability that the burner or control cannot provide. Combustion engineering expertise is available through your quality equipment suppliers and you should make use of this resource.





MANUFACTURED UNDER ONE
OR MORE OF THE FOLLOWING
PATENT NUMBERS:
3418062
3771944
4162890

MARK	DATE	REVISION

BLOOM ENGINEERING CO., INC
PITTSBURGH, PA.

GENERAL ARRANGEMENT OF
BLOOM HOT AIR LONOX BURNER

DATE
SCALE
DWN
APP'D
CHK.

SKX-B81-1

SHOP NOTES: ANGULAR TOL. ±1° 30'
ALL MACHINING TOLERANCE DIMENSIONS ARE
DECIMAL ±.005" | UNLESS OTHERWISE NOTED.
FRACTIONAL ±1/64" |
FABRICATION TOL ±1/8 ON OVERALL DIMENSIONS

ALL REFRACTORY SHAPES TO BE HELD WITHIN PLUS OR MINUS 5% OF OVERALL DIMENSIONS UNLESS SPECIFIED OTHERWISE

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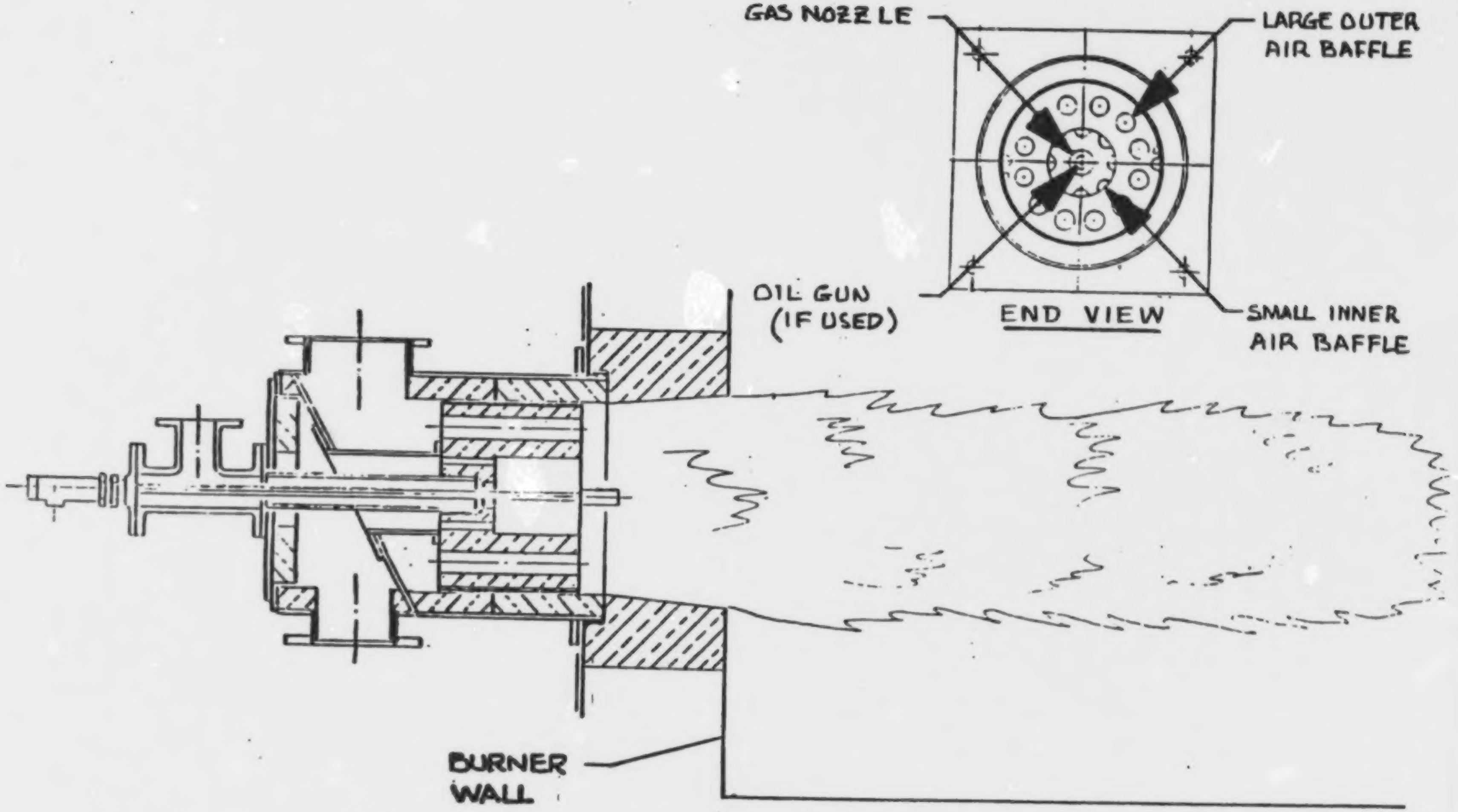
ENGINEERING SPECIFICATIONS

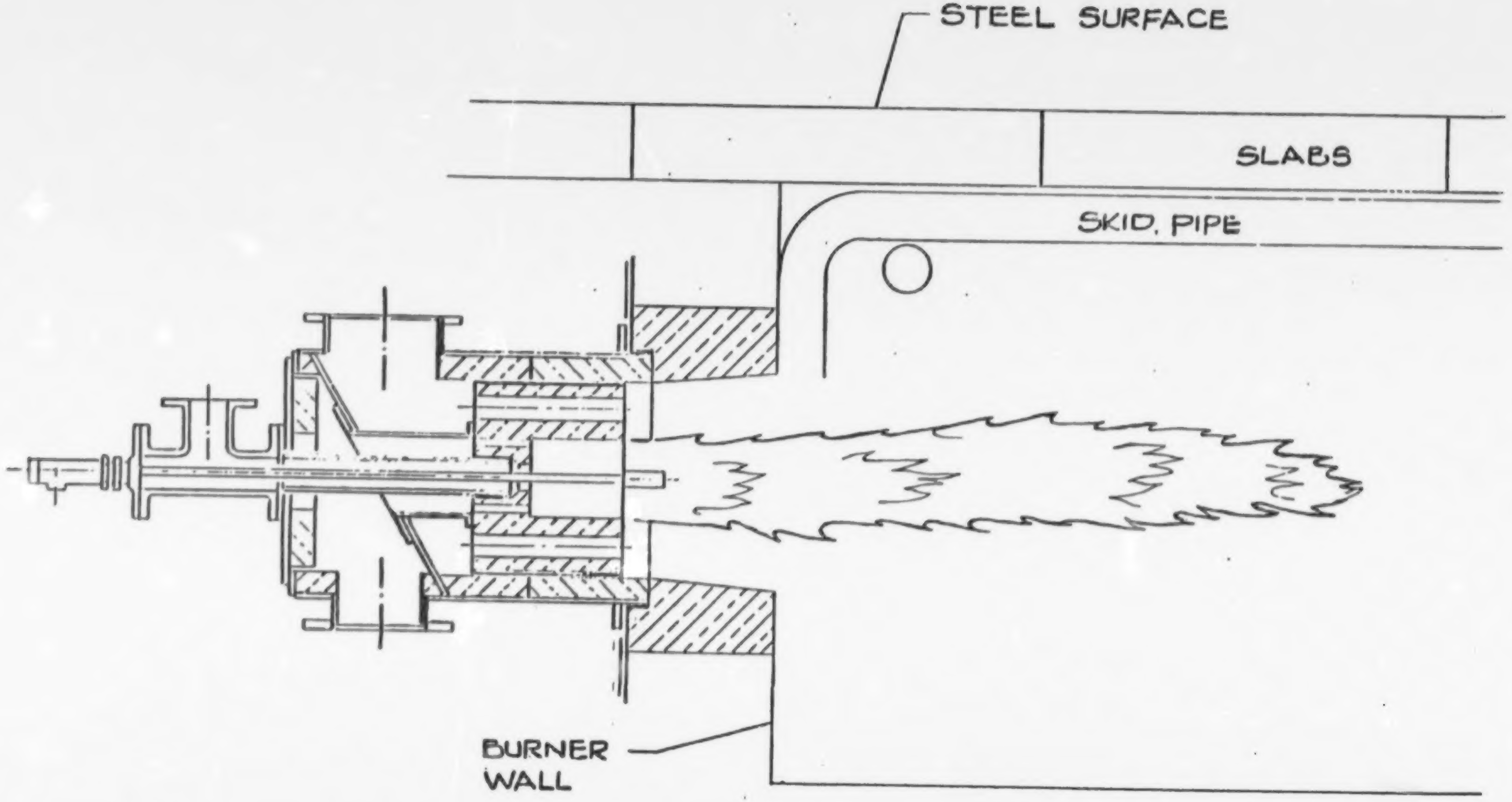
EXHIBIT 1


GAS NOZZLE
LARGE OUTER AIR BAFFLE
END VIEW
SMALL INNER AIR BAFFLE
OIL GUN (IF USED)
BURNER WALL

BLOOM 1170 BURNER

MAXIMUM FIRING TO 25 PERCENT

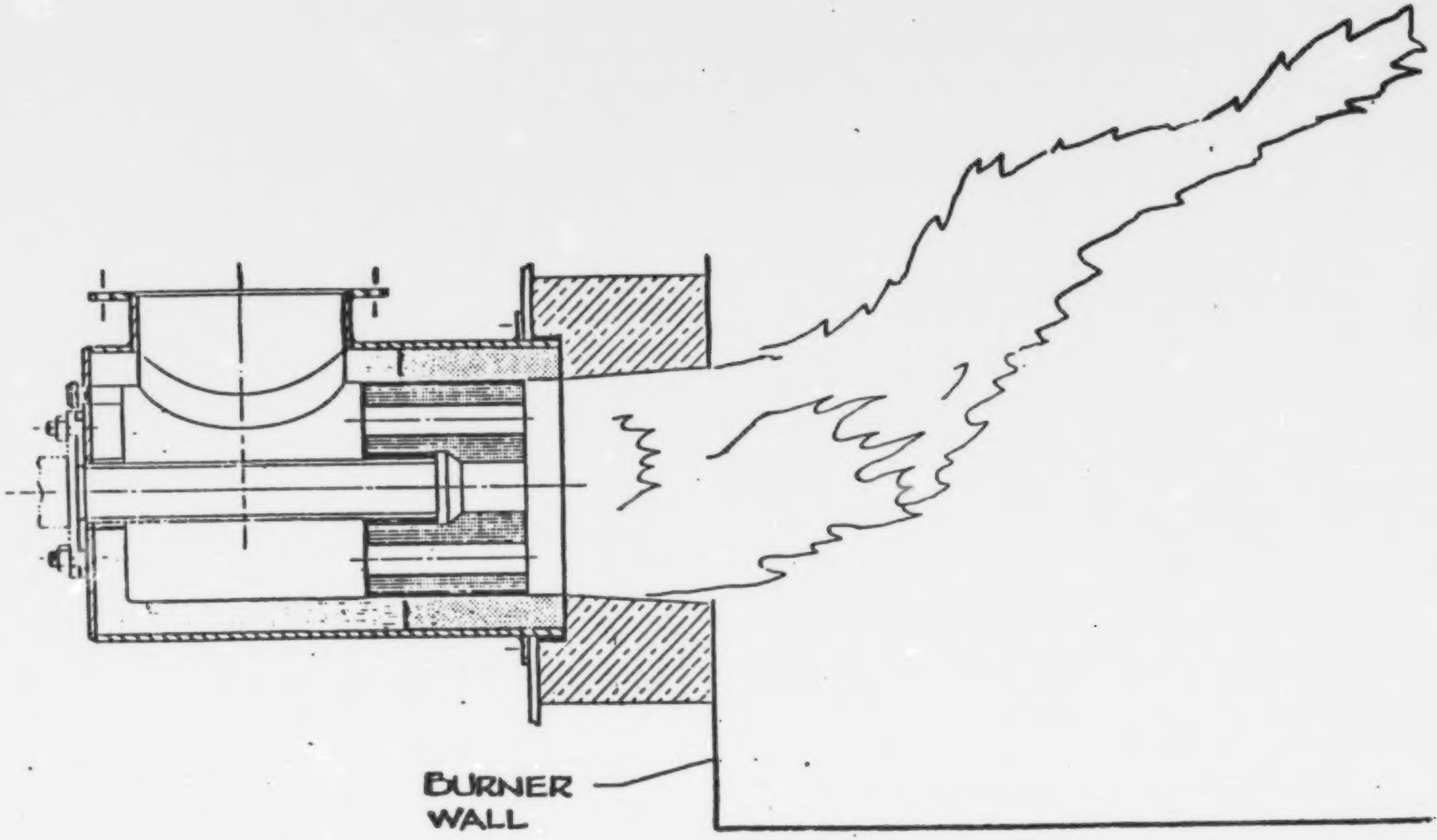
EXHIBIT 2


STEEL SURFACE
SLABS
SKID, PIPE
BURNER WALL

BLOOM 1170 BURNER

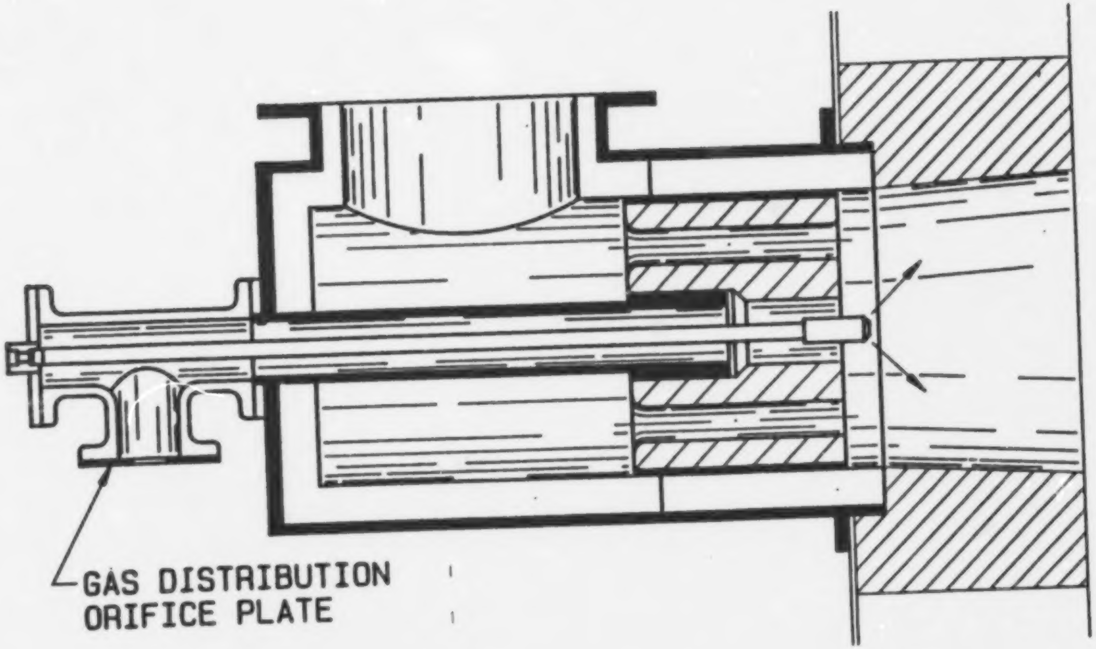
FIRING RATE 25 PERCENT TO 6 PERCENT

EXHIBIT 3


BURNER
WALL

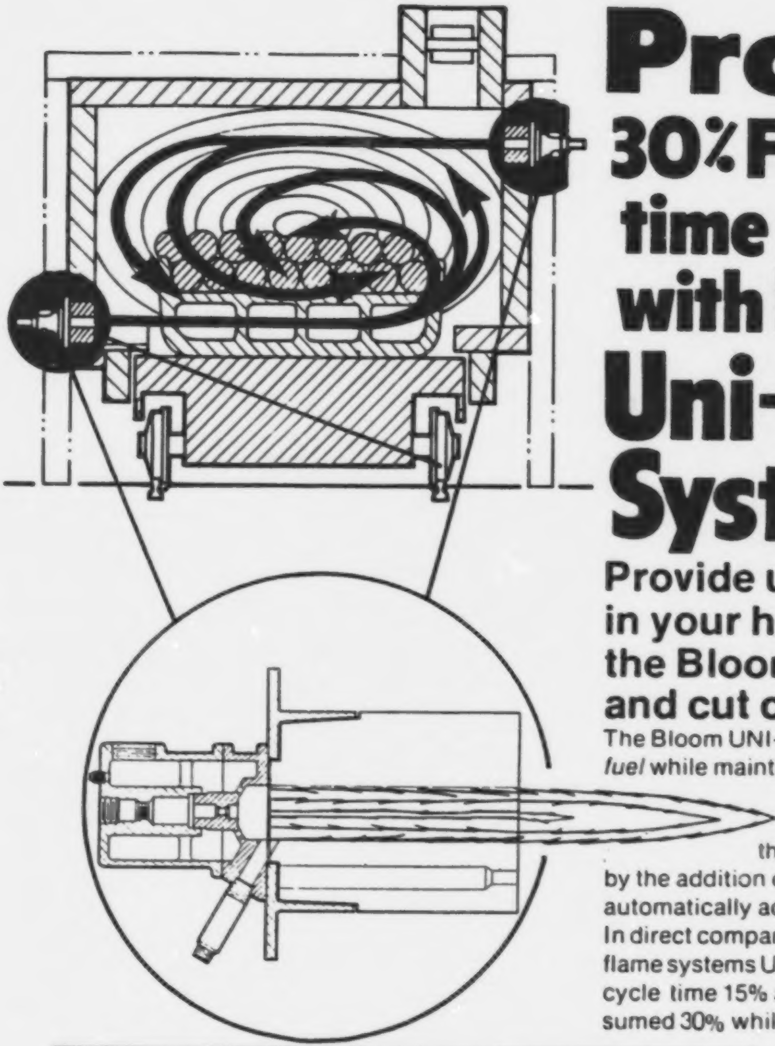
CONVENTIONAL BURNERS BELOW 25 PERCENT INPUT
WITH 10% EXCESS AIR

EXHIBIT 4


GAS DISTRIBUTION
ORIFICE PLATE

BLOOM 1200 SERIES FTR BURNERS
WITH HIGH TURNDOWN LANCE
SHORT FLAME

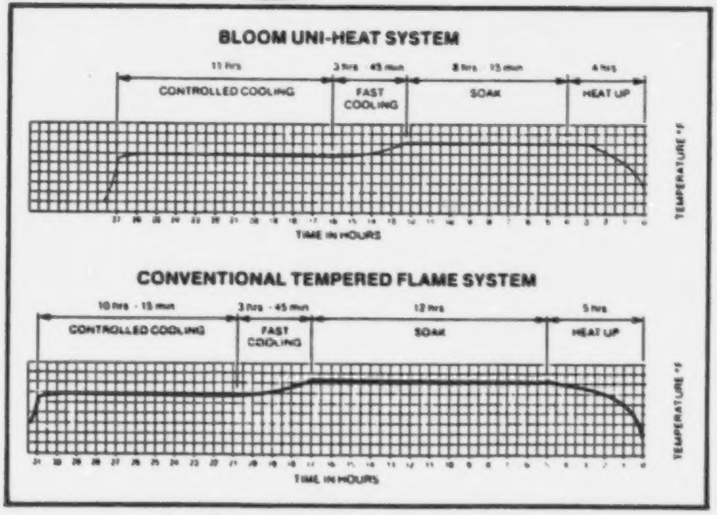
EXHIBIT 5



Proven
30% Fuel-15% Cycle time savings with Bloom
Uni-Heat™ System

Provide uniform temperature in your heat treat furnace with the Bloom Uni-Heat Burner and cut costs

The Bloom UNI-HEAT SYSTEM is *designed to save fuel* while maintaining uniform temperatures in your heat treat furnace. Its unique design maintains an even temperature throughout the full range of operation by the addition of high velocity air jets which are automatically activated under turndown conditions. In direct comparisons with conventional tempered flame systems UNI-HEAT has been proven to reduce cycle time 15% and reduce the amount of fuel consumed 30% while improving temperature uniformity and product quality of Car Type Heat Treat Furnaces during 1500°F annealing cycles. The Uni-Heat System is applicable to new or existing furnaces. For complete information call: 412/892-2121, or write Bloom Engineering Co., Inc., Horning & Curry Roads, Pittsburgh, PA 15236.



Furnace time comparisons: Bloom Uni-Heat System vs. conventional tempered flame system. Typical temperature vs. time curves for car type annealing furnace.

*Patent 4083677

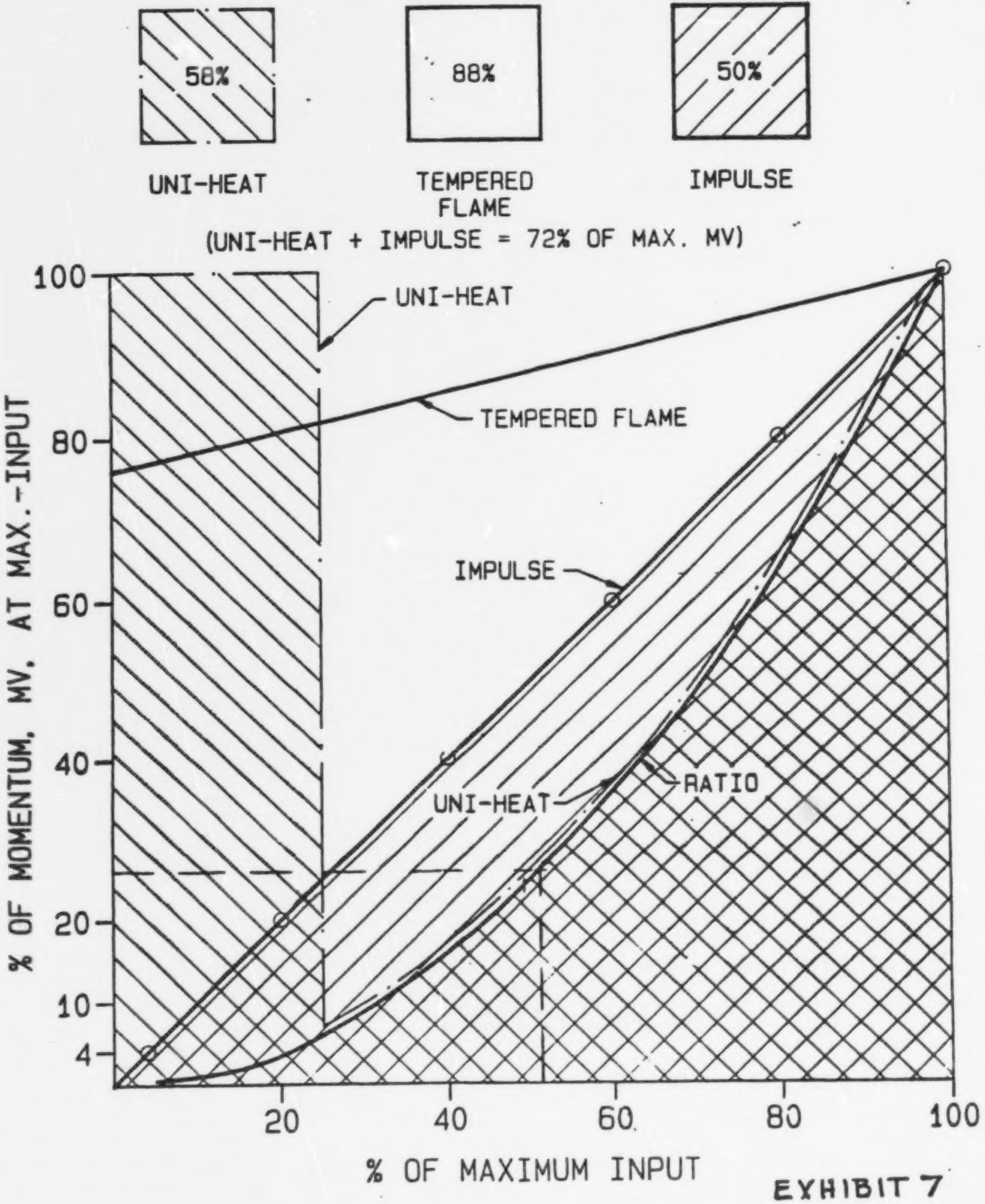
BLOOM ENGINEERING COMPANY, INC.

OFFICES Buffalo • Chicago • Cleveland • Philadelphia • Corte Madera, Calif. • Pittsburgh
SALES REPRESENTATIVES Dallas • Mexico City • Monterrey

EXHIBIT 6

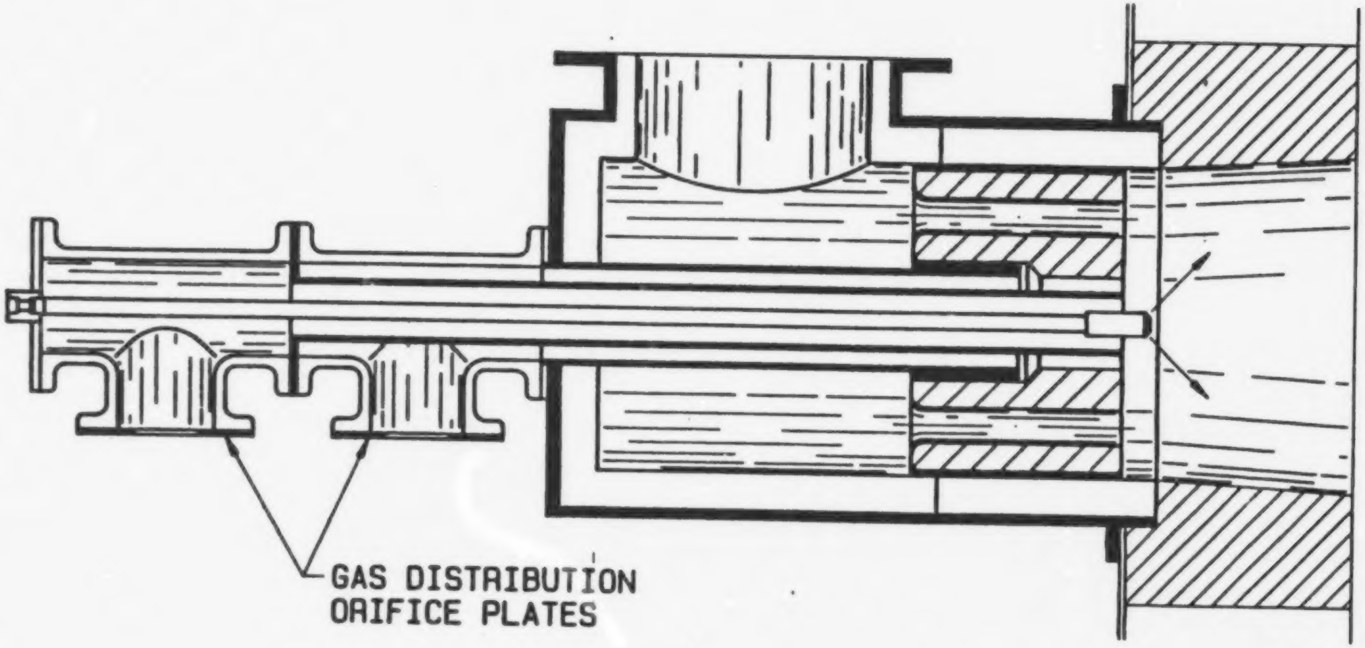
BLOOM ENGINEERING CO., INC.

COMPARISON - MOMENTUM LEVEL - HEAT TREATS FOR VARIOUS METHODS - WITH 10% EXCESS AIR % OF MAX. MOMENTUM - AVERAGE FOR CYCLE



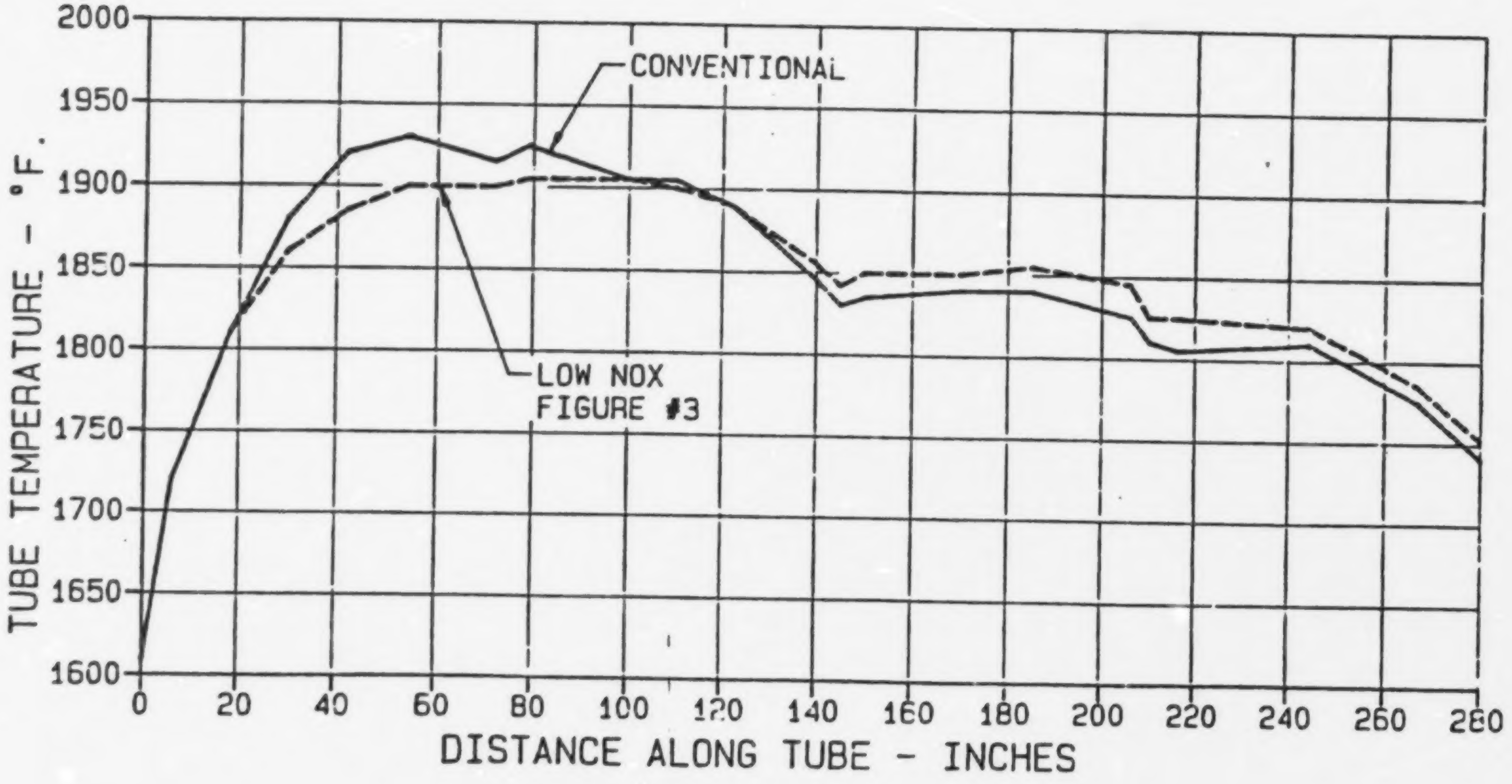
(UNI-HEAT + IMPULSE = 72% OF MAX. MV)

EXHIBIT 7


GAS DISTRIBUTION
ORIFICE PLATES

BLOOM 1200 SERIES ATR BURNERS
FUEL ENERGY
WITH HIGH TURNDOWN LANCE

EXHIBIT 8

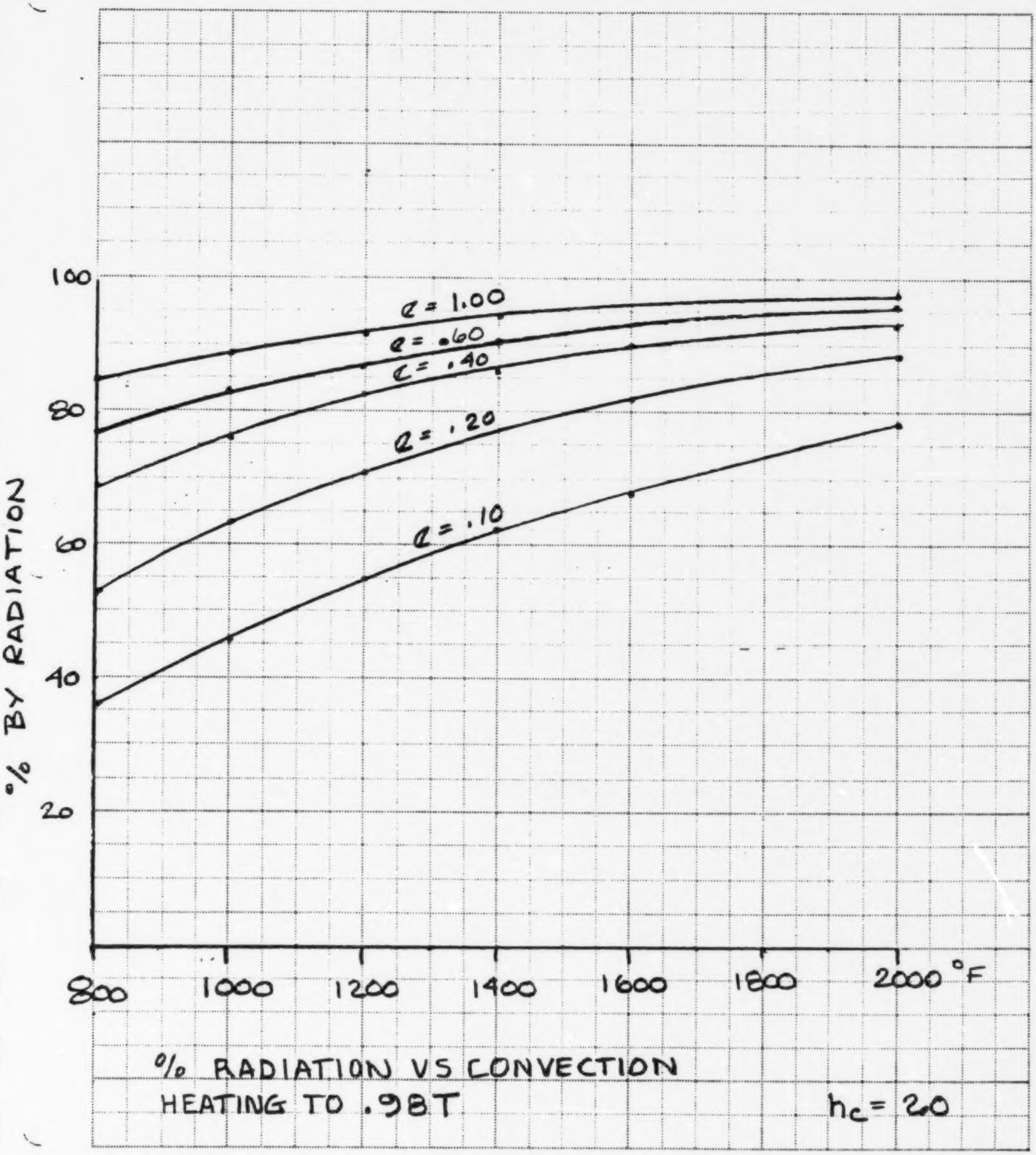

CONVENTIONAL
LOW NOX
FIGURE #3
TUBE TEMPERATURE - °F.
2000
1950
1900
1850
1800
1750
1700
1650
1600
0
20
40
60
80
100
120
140
160
180
200
220
240
260
280
DISTANCE ALONG TUBE - INCHES

EXHIBIT

HTR BURNER RADIATION IN COLD FURNACE

EXHIBIT 10

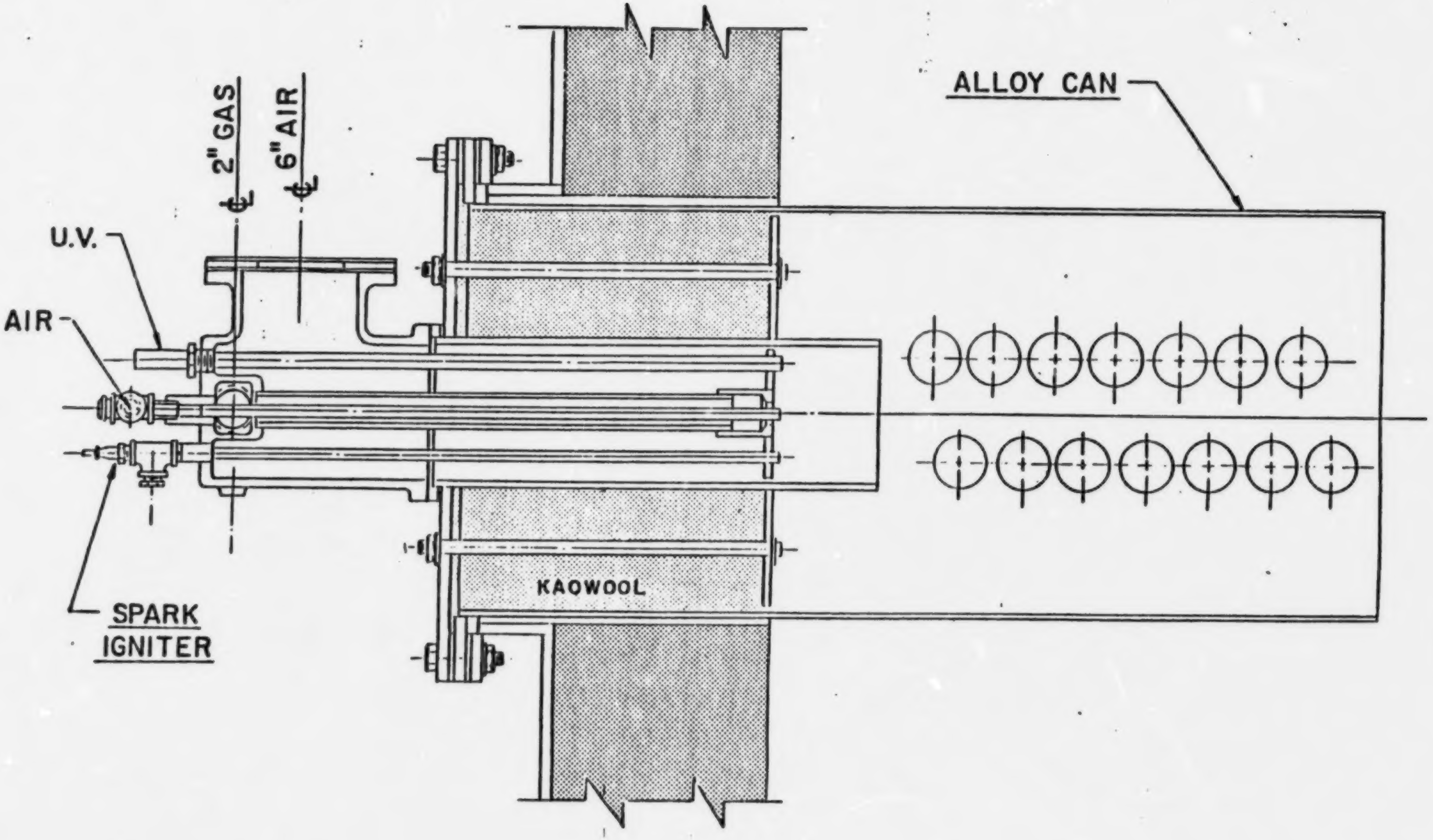
JOB NO. .... DWG. NO. .... DATE ....
SUBJECT .... BY ....



% RADIATION VS CONVECTION
HEATING TO .98T

$h_c = 20$

EXHIBIT 11


2" GAS
6" AIR
ALLOY CAN
U.V.
AIR
SPARK IGNITER
KAOWOOL

S-2300-040 BURNER ARRG'T.
FOR HOMOGENIZING FURNACE
REFRACTORY PORT BLOCK NOT REQ'D.

EXHIBIT 12

SECTION

XVI

HAZARDOUS WASTE INCINERATION

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B.C.
13 - 14 April, 1988

by John Young
John Zinc Company

INTRODUCTION

John Zink Company owns and operates a Research Facility for testing combustion systems designed to treat hazardous wastes.

The PCS (Primary Combustion System) consists of various process modules (i.e., combustion, heat recovery, air pollution control) coupled together in a suitable arrangement to treat a specific test waste. All process modules are sized for a firing rate of 3.0 MM BtuH.

Testing follows an operating temperature-residence time parameter matrix which is set prior to the test by using in-house design criteria for a similar waste(s). In the event that the selected operating parameters are inadequate, the PCS will not operate within the requirements set forth in 40 CFR 264.343, which sets a minimum DRE of 99.99% for POHC(s), and maximum emission rates for hydrochloric acid and particulate matter.

The use of an incinerator/afterburner (I/A) in series with the PCS treating all the flue gas from the PCS will cause the overall combustion process to achieve a DRE of at least 99.99%. The combination of a PCS and I/A for permitting purposes is called a Test Combustion System.

Candidate test materials can be virtually any chemical and/or combination of chemicals. However, it was deemed inappropriate to test with materials containing dioxins, PCB is at concentrations greater that 50 ppm or EP toxic metal concentrations greater than those listed in Table 1 of 40 CFR 261.24. Also omitted are chemicals having a lower heat of combustion less than carbon tetrachloride, which was the trial burn waste. The I/A by itself, independently of the performance of a PCS preceding it, must have the capability to achieve a DRE of at least 99.99% for the Test Combustion System to always be in compliance. Thus, only the I/A is subject to RCRA regulation.

EPA requires that the owner/operator of an incinerator treating hazardous waste prove that the incinerator meets the previously stated criteria. This is accomplished by conducting a Trial Burn. An acceptable Trial Burn Plan must be submitted to the EPA before a Trial Burn can be conducted. The results of the Trial Burn and any variance from the Trial Burn Plan are documented in the Trial Burn Report.

This paper is divided into three sections. Section I briefly describes the test equipment available at the John Zink Research Facility, Section II the Trial Burn Plan, and Section III, the Trial Burn Report.

The reader will note that control of emissions rates for hydrochloric acid and particulate matter are provided by the PCS. The I/A is only used to insure destruction of POHC(s). Discussion of the integration of hydrochloric acid and particulate matter control into the Part B section of RCRA permitting activity is outside the scope of this paper.

RESEARCH FACILITY

Available at the Research Facility are sufficient process modules to assemble a semi-commercial version of virtually any hazardous waste system. These modules, organized according to function, are listed in Table I.

TABLE I AVAILABLE PROCESS MODULES

Designation	Description	Comments
	Primary Combustion	
0-1	Vertical down-fired	Liquid, gas waste
0-2	Vertical down-fired	Liquid, gas waste
0-3	Horizontal side-fired	Liquid, gas waste
0-4A	Vertical up-fired	Liquid, gas waste
0-4B	Vertical down-fired	Liquid, gas waste
0-5	Horizontal cyclonic (hotwall)	Small solids waste
0-6	Vertical up-fired	Liquid, gas waste
0-7	Horizontal cyclonic (coldwall)	Small solids waste
0-8	Horizontal (rotating)	All wastes
	Flue Gas Conditioning	
T-1	Vertical down-flow	Cool flue gas
T-2	Vertical up-flow	Cool flue gas
T-3	Vertical down-flow	Saturated flue gas
T-4	Horizontal cross-flow	Saturated flue gas
	Heat Recovery	
M-1	Water-tube boiler	Clean service
M-2	Fire-tube boiler	Slightly dirty service
	Acid Removal	
A-1	Packed-bed absorber	
A-2	Venturi scrubber (fixed throat)	
A-3	Venturi scrubber (adjustable throat)	
	Particulate Removal	
D-1	Baghouse (4-bags)	
A-2	Venturi scrubber (fixed throat)	
A-3	Venturi scrubber (adjustable throat)	
	Burners	
B-1	Low-to-medium intensity	
B-2	High intensity	

Many burner/waste gun combinations are available or can be custom manufactured for any test waste.

Four general classes of primary combustion systems can be made available for testing. These classes, organized according to system function, are listed in Table II.

TABLE II PRIMARY COMBUSTION SYSTEM FUNCTION

System	Function Performed
Type 1	Combustion
Type 2	Combustion HCl removal
Type 3	Combustion Particulate removal
Type 4	Combustion HCl removal Particulate removal

The Type 1 incineration process is used to treat hazardous wastes that do not require post incineration treatment. Figure 1 contains four diagrams, one for each of the four Type 1 Primary Combustion system configurations that can be tested.

The Type 2 incineration process is used to treat hazardous wastes that require post incineration treatment for HCl removal. Figure 2 contains two block diagrams, one for each Type 2 Primary Combustion System configuration that can be tested.

The Type 3 incineration process is used to treat hazardous wastes that require post incineration treatment for particulate removal. Figure 3 contains three block diagrams, one for each Type 3 Primary Combustion System configuration that can be tested.

The Type 4 incineration system is used to treat hazardous wastes that require post incineration treatment for HCl and particulate removal. Figure 4 contains three block diagrams, one for each Type 4 Primary Combustion System Configuration that can be tested.

Note, that all configurations shown are for testing with hazardous wastes. All test combustion systems are fitted with an I/A providing final treatment.

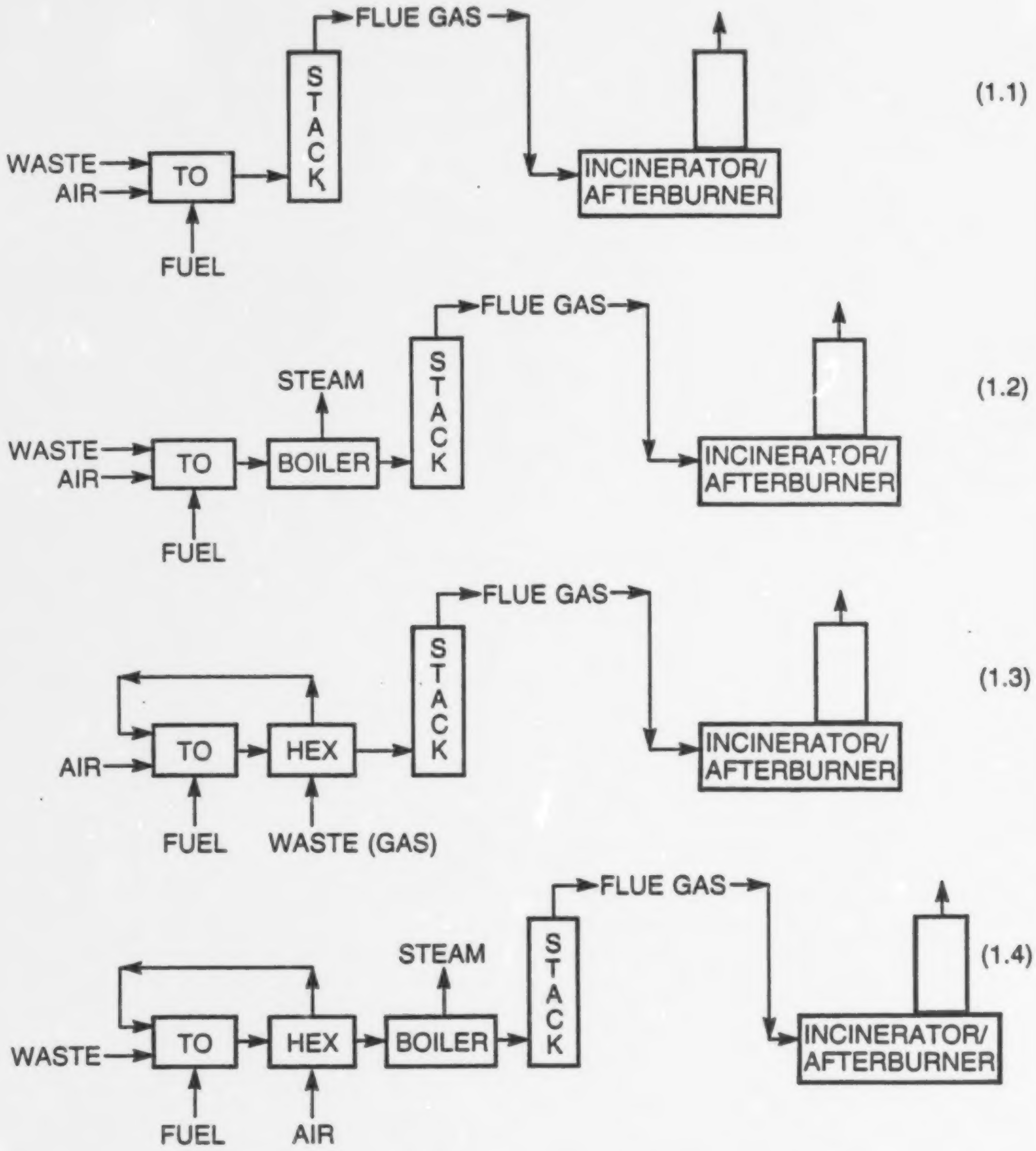

FLUE GAS
STACK
WASTE
AIR
TO
FUEL
INCINERATOR/ AFTERBURNER
(1.1)
FLUE GAS
STEAM
WASTE
AIR
TO
BOILER
STACK
FUEL
INCINERATOR/ AFTERBURNER
(1.2)
FLUE GAS
AIR
TO
HEX
STACK
FUEL
WASTE (GAS)
INCINERATOR/ AFTERBURNER
(1.3)
FLUE GAS
STEAM
WASTE
TO
HEX
BOILER
STACK
FUEL
AIR
INCINERATOR/ AFTERBURNER
(1.4)

FIGURE 1 TYPE 1 INCINERATION PROCESS

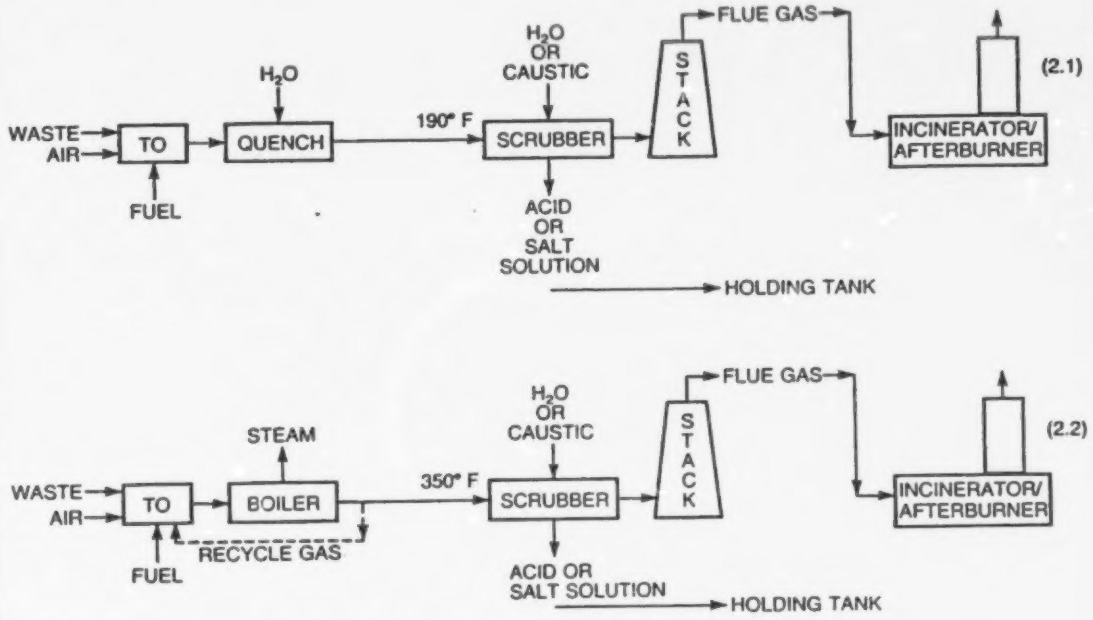

WASTE
AIR
TO
FUEL
H2O
QUENCH
190° F
H2O OR CAUSTIC
SCRUBBER
ACID OR SALT SOLUTION
HOLDING TANK
STACK
FLUE GAS
INCINERATOR/ AFTERBURNER
(2.1)
WASTE
AIR
TO
FUEL
STEAM
BOILER
RECYCLE GAS
350° F
H2O OR CAUSTIC
SCRUBBER
ACID OR SALT SOLUTION
HOLDING TANK
STACK
FLUE GAS
INCINERATOR/ AFTERBURNER
(2.2)

FIGURE 2 TYPE 2 INCINERATION PROCESS

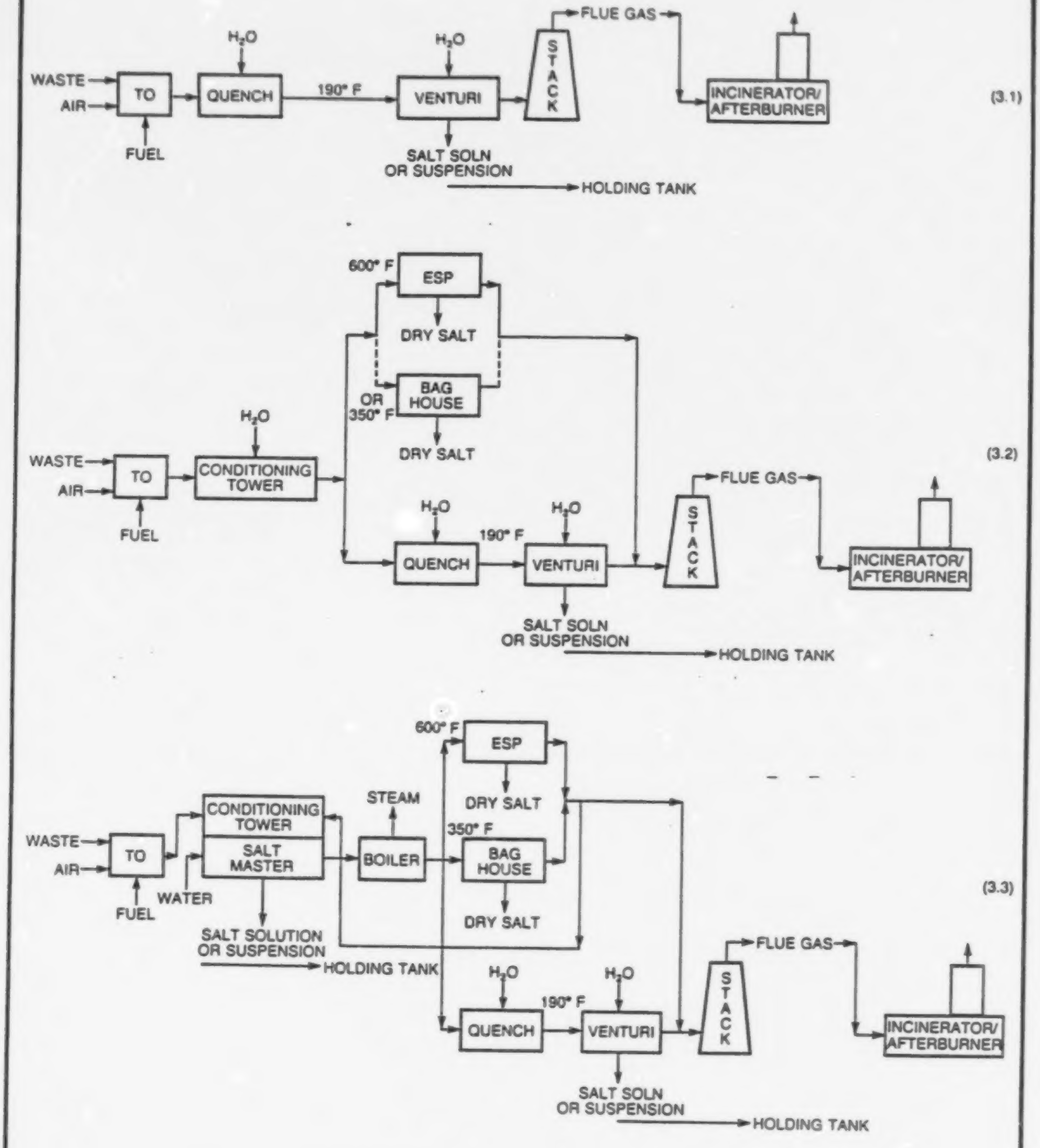

WASTE
AIR
TO
FUEL
H$_2$O
QUENCH
190° F
H$_2$O
VENTURI
STACK
FLUE GAS
INCINERATOR/ AFTERBURNER
SALT SOLN OR SUSPENSION
HOLDING TANK
(3.1)
600° F
ESP
DRY SALT
OR 350° F
BAG HOUSE
DRY SALT
H$_2$O
WASTE
AIR
TO
FUEL
CONDITIONING TOWER
H$_2$O
QUENCH
190° F
H$_2$O
VENTURI
STACK
FLUE GAS
INCINERATOR/ AFTERBURNER
SALT SOLN OR SUSPENSION
HOLDING TANK
(3.2)
600° F
ESP
DRY SALT
STEAM
CONDITIONING TOWER
SALT MASTER
BOILER
350° F
BAG HOUSE
DRY SALT
WASTE
AIR
TO
FUEL
WATER
SALT SOLUTION OR SUSPENSION
HOLDING TANK
H$_2$O
QUENCH
190° F
H$_2$O
VENTURI
STACK
FLUE GAS
INCINERATOR/ AFTERBURNER
SALT SOLN OR SUSPENSION
HOLDING TANK
(3.3)

FIGURE 3 TYPE 3 INCINERATION PROCESS

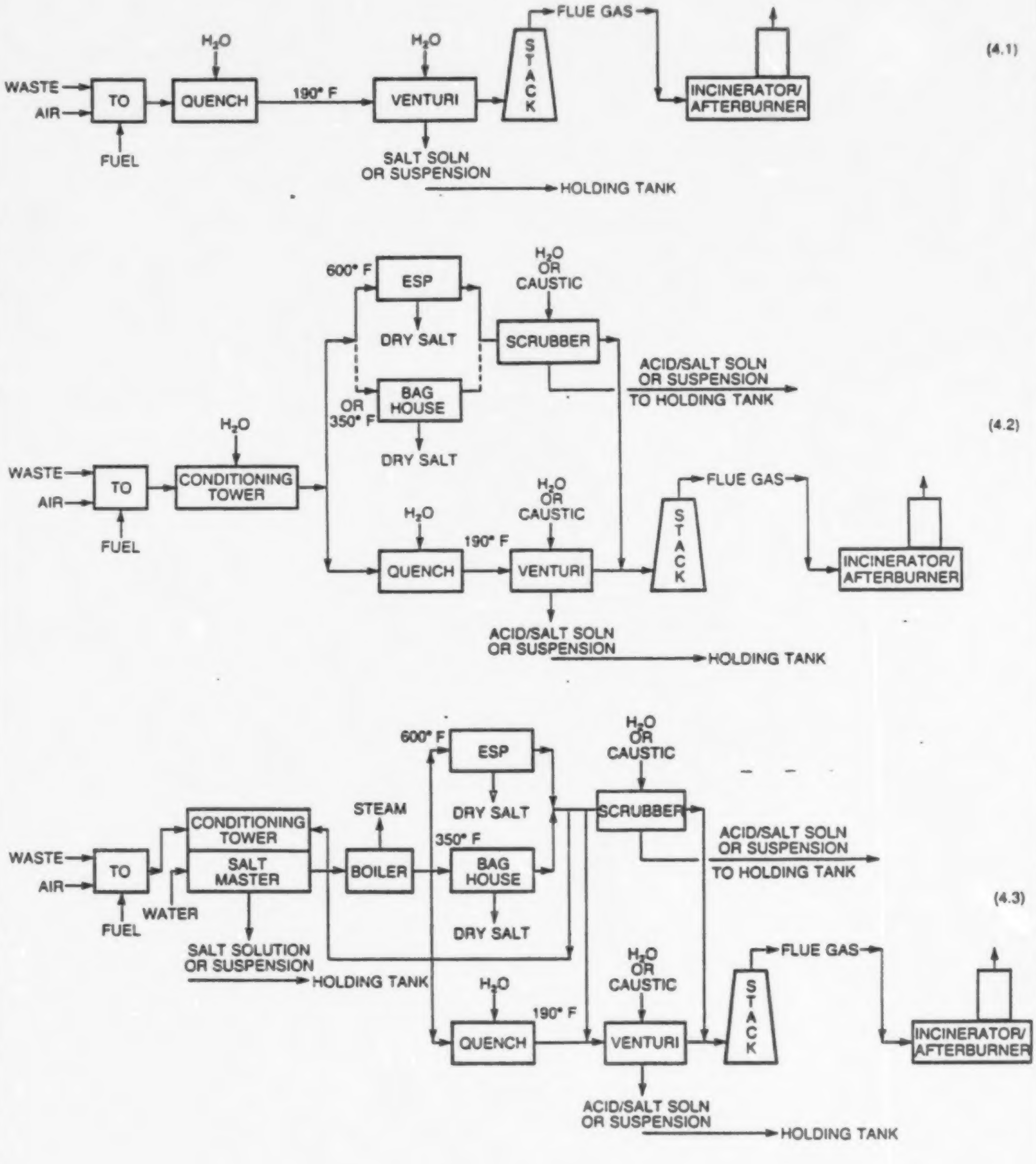

WASTE
AIR
TO
FUEL
H₂O
QUENCH
190° F
H₂O
VENTURI
SALT SOLN OR SUSPENSION
HOLDING TANK
STACK
FLUE GAS
INCINERATOR/ AFTERBURNER
(4.1)
WASTE
AIR
TO
FUEL
H₂O
CONDITIONING TOWER
600° F
ESP
DRY SALT
OR 350° F
BAG HOUSE
DRY SALT
H₂O OR CAUSTIC
SCRUBBER
ACID/SALT SOLN OR SUSPENSION
TO HOLDING TANK
(4.2)
H₂O
QUENCH
190° F
H₂O OR CAUSTIC
VENTURI
ACID/SALT SOLN OR SUSPENSION
HOLDING TANK
STACK
FLUE GAS
INCINERATOR/ AFTERBURNER
WASTE
AIR
TO
FUEL
WATER
CONDITIONING TOWER
SALT MASTER
SALT SOLUTION OR SUSPENSION
HOLDING TANK
STEAM
BOILER
600° F
ESP
DRY SALT
350° F
BAG HOUSE
DRY SALT
H₂O OR CAUSTIC
SCRUBBER
ACID/SALT SOLN OR SUSPENSION
TO HOLDING TANK
(4.3)
H₂O
QUENCH
190° F
H₂O OR CAUSTIC
VENTURI
ACID/SALT SOLN OR SUSPENSION
HOLDING TANK
STACK
FLUE GAS
INCINERATOR/ AFTERBURNER

FIGURE 4 TYPE 4 INCINERATION PROCESS

TRIAL BURN PLAN

INCINERATOR/AFTERBURNER DESIGN

Combustion tests are conducted with the PCS fired at a rate of 3.0 MM BtuH (lower heating value basis, Lhv) which includes both the auxiliary fuel and the combustion energy from the waste. A "worst case" design for the I/A is that in which methane (CH_4) represents an exothermic hazardous test material with combustor temperature controlled by excess air and the products of combustion are quenched to saturation by direct contact with water. "Worst case" design means a maximum flue gas flow rate is being supplied to the I/A requiring a maximum amount of auxiliary fuel firing for thermal treatment.

CH_4 can be demonstrated as a "worst case" design by performing combustion calculations with two compounds selected at random: Ethylene Oxide (C_2H_4O) and Ethylene Glycol ($C_2H_6O_2$), along with methane, which have Lhv(s) of 11481, 7257, and 21284 Btu/lb respectively. Table III lists the results of these calculations for an operating temperature of 1600°F, controlled by excess air.

TABLE III - RESULTS OF COMBUSTION CALCULATIONS

Compound	Lhv^{60}, Btu/lb	Flue Gas, Moles/hr
C_2H_4O	11,481	250.7
$C_2H_6O_2$	7,257	249.0
CH_4	21,284	251.2

These results indicate that the flue gas rate increases as combustion energy increases. However, all values are essentially the same. Thus, CH_4 is a valid representative of a test material in the design of the I/A to treat flue gas produced by a PCS.

An operating temperature of 1600°F was selected as a "worst case" temperature because virtually all testing is performed at an operating temperature exceeding 1600°F. Operating at 1600°F requires a maximum amount of air for temperature control, which means a maximum amount of flue gas is produced for thermal treatment by the I/A.

The rate of the flue gas products resulting from the combustion of 3.0 MM BtuH of CH_4 at a combustor module operating temperature of 1600°F, controlled by excess air is, listed in Table IV.

TABLE IV 3.0 MM BtuH COMBUSTION PRODUCTS

Component	Moles/hr
CO_2	8.79
H_2O	17.58
O_2	33.30
N_2	191.52
	Sum = 251.2 moles/hr

The flue gas can be ducted to the I/A at 1600°F or cooled first by a heat recovery boiler and/or direct contact with water. Use of a boiler will cool the flue gas to approximately 350°F. Direct contact with water will cool the flue gas to its saturation temperature, which is approximately 170°F. The "worst case" mass rate for the I/A is when the flue gas is cooled to saturation. This case requires a maximum amount of I/A auxiliary fuel, which means a maximum internal volume for a selected residence time.

The rate of the saturated flue gas products being supplied to the I/A at a saturation temperature of 170°F by the PCS is listed in Table V.

TABLE V - PCS COMBUSTION PRODUCTS

Component	Moles/Hr	Lb/Hr
CO_2	8.79	386.8
H_2O	158.98	2864.8
O_2	33.30	1065.6
N_2	191.52	5362.6
	Sum = 392.59 moles/hr	9679.8 lb/hr

A conservative I/A is one which offers 1.0 second of residence time at an operating temperature of 1800°F.

Figure 5 is a schematic of the I/A. It consists of a horizontal incinerator fitted with a double-pipe flue gas-to-combustion gas preheater and a vent stack. The PCS flue gas enters the preheater and flows co-current with the I/A flue gas. It then exits the preheater at a temperature of approximately 300°F and enters the incinerator in a tangential fashion to create maximum turbulence.

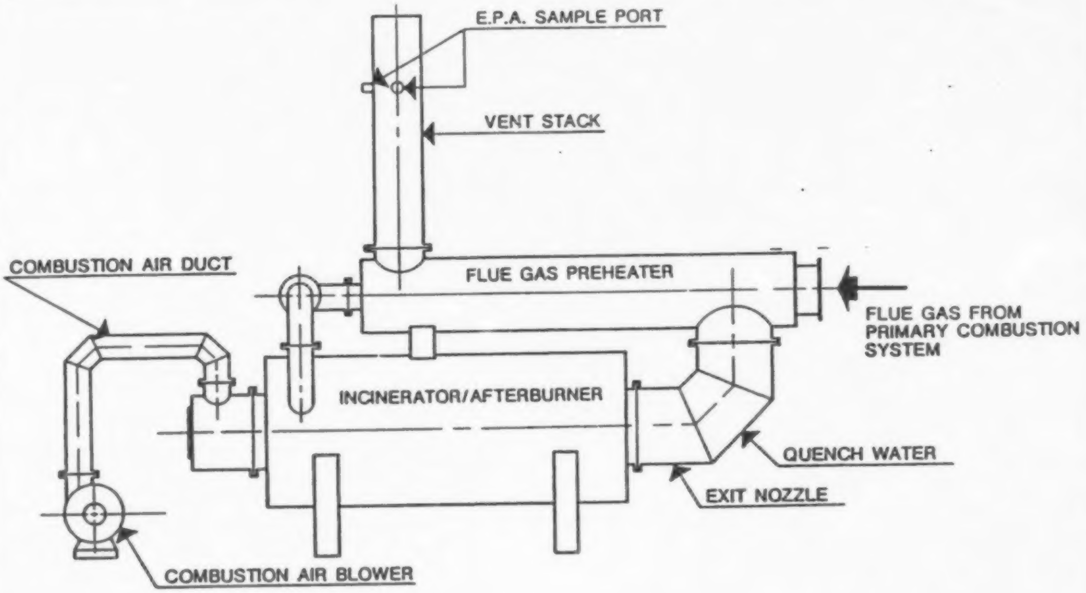

E.P.A. SAMPLE PORT
VENT STACK
COMBUSTION AIR DUCT
FLUE GAS PREHEATER
FLUE GAS FROM PRIMARY COMBUSTION SYSTEM
INCINERATOR/AFTERBURNER
QUENCH WATER
EXIT NOZZLE
COMBUSTION AIR BLOWER

FIGURE 5 INCINERATOR/AFTERBURNER

The I/A auxiliary fuel rate required to heat the flue gas to 1800°F, assuming that it does not have any heating value, is 9.982 MM Btu/hr. This is based on firing the auxiliary fuel with 10% excess combustion air.

The resulting combustion products are listed in Table VI.

TABLE VI - I/A COMBUSTION PRODUCTS

Component	PCS		I/A	Totals
CO_2	8.79	+	29.2	38.0
H_2O	158.98	+	58.4	217.4
O_2	33.30	+	5.8	39.1
N_2	191.52	+	242.4	433.9
	392.59	+	335.8	728.4

The concentration of O_2 on a dry basis of 7.65%.

The required combustion volume, (V_c) for a residence time, (R_t) of 1.0 second is determined as follows:

V_c, ft^3 = flue gas rate (ACFS) X 1.0 sec. (R_t)

Where ACFS = actual ft^3/sec.

$$\text{Thus, ACFS} = \frac{728.4 \text{ moles/hr}}{3600 \text{ sec/hr}} \text{ X } 379 \text{ ft}^3\text{/Mole X } \frac{2260°\text{R X } 14.7 \text{ psia}}{520°\text{R X } 14.7 \text{ psia}}$$

$$= 333.3 \text{ ft}^3\text{/sec. at } 1800°\text{F.}$$

The I/A has an internal volume of 359 ft^3. This equates to a residence time of 1.08 seconds for the "worst case" conditions. Any temperature increase in the flue gas caused by the heat exchanger or a higher temperature flue gas from the PCS will reduce the I/A's auxiliary fuel requirement, which means additional residence time will be available for treatment.

Figure 6 is a schematic displaying the energy and mass balances for the "worst case" conditions.

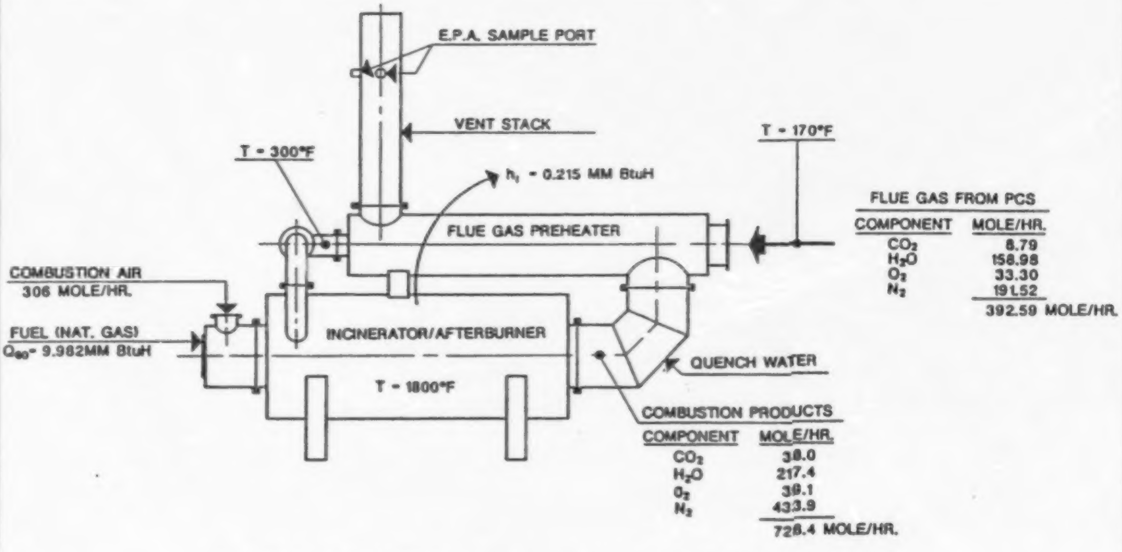

E.P.A. SAMPLE PORT
VENT STACK
T - 170°F
T - 300°F
h_l - 0.215 MM BtuH
FLUE GAS FROM PCS
COMPONENT MOLE/HR.
CO2 8.79
H2O 158.98
O2 33.30
N2 191.52
392.59 MOLE/HR.
FLUE GAS PREHEATER
COMBUSTION AIR
306 MOLE/HR.
FUEL (NAT. GAS)
Q80- 9.982MM BtuH
INCINERATOR/AFTERBURNER
T - 1800°F
QUENCH WATER
COMBUSTION PRODUCTS
COMPONENT MOLE/HR.
CO2 38.0
H2O 217.4
O2 39.1
N2 433.9
728.4 MOLE/HR.

FIGURE 6 INCINERATOR/AFTERBURNER HEAT AND MASS BALANCE

INCINERATOR/AFTERBURNER EQUIPMENT DESCRIPTION

The I/A is cylindrical in shape and is positioned horizontally. It is a nominal 6′-0″ O.D. x 15′-0″ in length. The front wall is fitted with a DBY-0-10 forced draft gas burner rated at 10 MM BtuH. The rear wall is fitted with a 2′-0″ I.D. flue gas exit nozzle. The flue gas from the PCS enters the I/A via a 1′-0″ I.D. nozzle located 1′-0″ from the front wall, and is oriented to provide tangential entry. All refractory materials will allow safe operation at temperatures exceeding 2000°F.

The flue gas exit nozzle connects to a 24″ I.D. unlined duct which serves to convey these gases to the vent stack and form the outer heat transfer surface for the PCS flue gas to I/A flue gas double-pipe type flue gas preheater. The vent stack is 11′-0″ in length and is fitted with two 3″ NPT sample ports, positioned 90° apart. These ports are located 7′-0″ from the center line of the vent stack to the preheater flange. The vent stack L/D prior to the port is 4.0 and after the port is 1.5.

The I/A exit nozzle is fitted with a coupling to accept a water tempering nozzle sized to lower the flue gas temperature from 1800°F to 1600°F to prolong exchanger life.

Sufficient thermocouple connections and sample ports are included to control the measure I/A operation.

INCINERATOR/AFTERBURNER TRIAL BURN CONFIGURATION

Figure 7 is a schematic of the trial burn configuration. It consists of a PCS, hazardous waste feed system and I/A.

The PCS is shown schematically in Figure 8 to consist of a (down-fired) thermal oxidizer fitted with a natural gas burner, quench section, quench pot, adjustable venturi, separator, interconnecting duct and combustion air supply.

The parameters to be measured and the approximate location of the sample points are also shown schematically in Figure 8. Table VII describes in more detail the parameters to be measured and the type of instrumentation to be used.

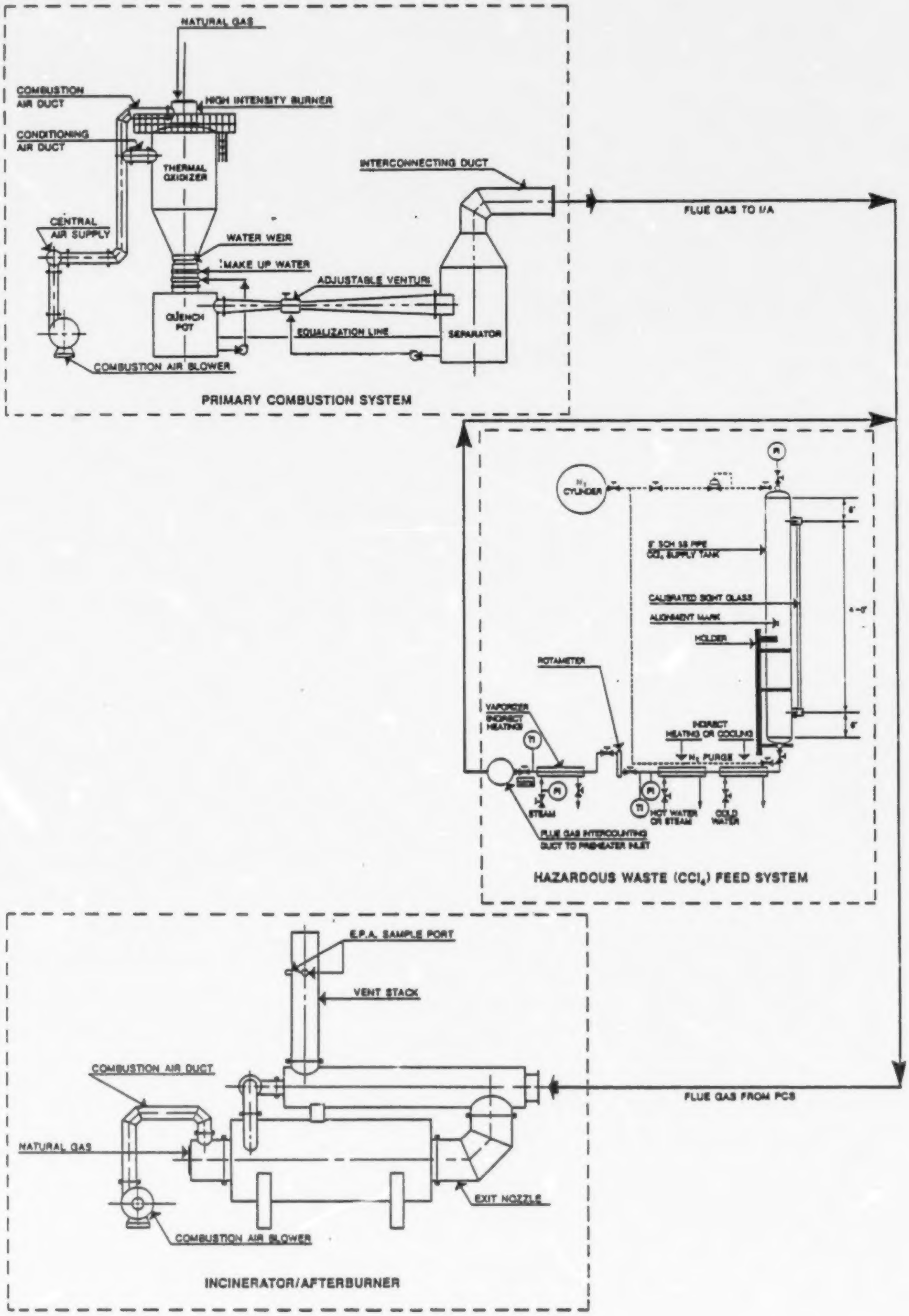

NATURAL GAS
COMBUSTION AIR DUCT
HIGH INTENSITY BURNER
CONDITIONING AIR DUCT
THERMAL OXIDIZER
INTERCONNECTING DUCT
CENTRAL AIR SUPPLY
WATER WEIR
MAKE UP WATER
ADJUSTABLE VENTURI
QUENCH POT
EQUALIZATION LINE
SEPARATOR
COMBUSTION AIR BLOWER
PRIMARY COMBUSTION SYSTEM
FLUE GAS TO I/A
PI
N_2 CYLINDER
6"
5" SCH 38 PIPE
CCl_4 SUPPLY TANK
CALIBRATED SIGHT GLASS
4'-0"
ALIGNMENT MARK
HOLDER
ROTAMETER
VAPORIZER INDIRECT HEATING
TI
INDIRECT HEATING OR COOLING
N_2 PURGE
6"
PI
STEAM
PI
TI
HOT WATER OR STEAM
COLD WATER
FLUE GAS INTERCOUNTING DUCT TO PREHEATER INLET
HAZARDOUS WASTE (CCl_4) FEED SYSTEM
E.P.A. SAMPLE PORT
VENT STACK
COMBUSTION AIR DUCT
FLUE GAS FROM PCS
NATURAL GAS
EXIT NOZZLE
COMBUSTION AIR BLOWER
INCINERATOR/AFTERBURNER

FIGURE 7 TRIAL BURN CONFIGURATION

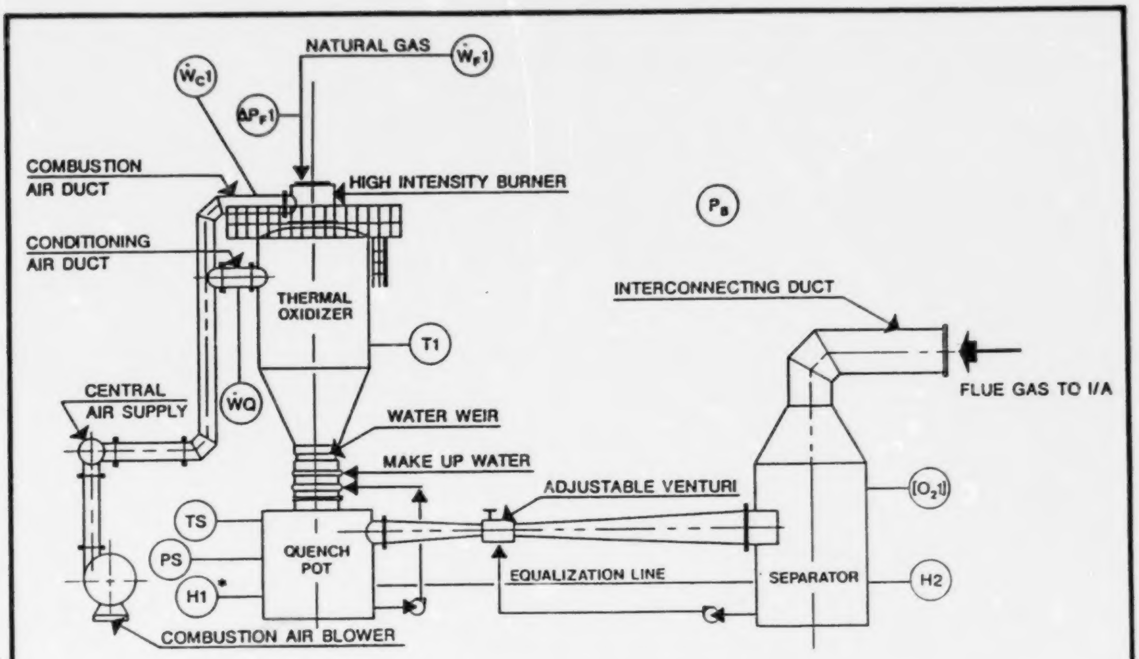

NATURAL GAS
Ẇ$_F$1
Ẇ$_C$1
ΔP$_F$1
COMBUSTION AIR DUCT
HIGH INTENSITY BURNER
P$_B$
CONDITIONING AIR DUCT
THERMAL OXIDIZER
INTERCONNECTING DUCT
T1
FLUE GAS TO I/A
CENTRAL AIR SUPPLY
ẆQ
WATER WEIR
MAKE UP WATER
ADJUSTABLE VENTURI
[O$_2$1]
TS
PS
H1
QUENCH POT
EQUALIZATION LINE
SEPARATOR
H2
COMBUSTION AIR BLOWER

FIGURE 8 PROCESS COMBUSTION SYSTEM

TABLE VII - PCS MEASURED PARAMETERS AND INSTRUMENTATION

Parameter	Instrument
Natural gas flow rate, W_F1	Rotameter Model 1110 manufactured by Brooks, Inc. with a capacity of 4.0 MM BtuH
Oxygen Concentration (O_21)	Cell type meter model 980 manufactured by Teledyne.
Combustor temperature, T1	Type K thermocouple manufactured by Westronics, Inc.
Saturation temperature, TS	Type K thermocouple manufactured by Westronics, Inc.
Saturation pressure, PS	Pressure gauge manufactured by Ashcroft or equivalent with a capacity of 0 - 100" W.C.

The I/A is shown schematically in Figure 9. Included in this figure are the parameters to be measured and the approximate location of the sample points. Table VIII describes in more detail the parameters to be measured and the type of instrumentation to be used.

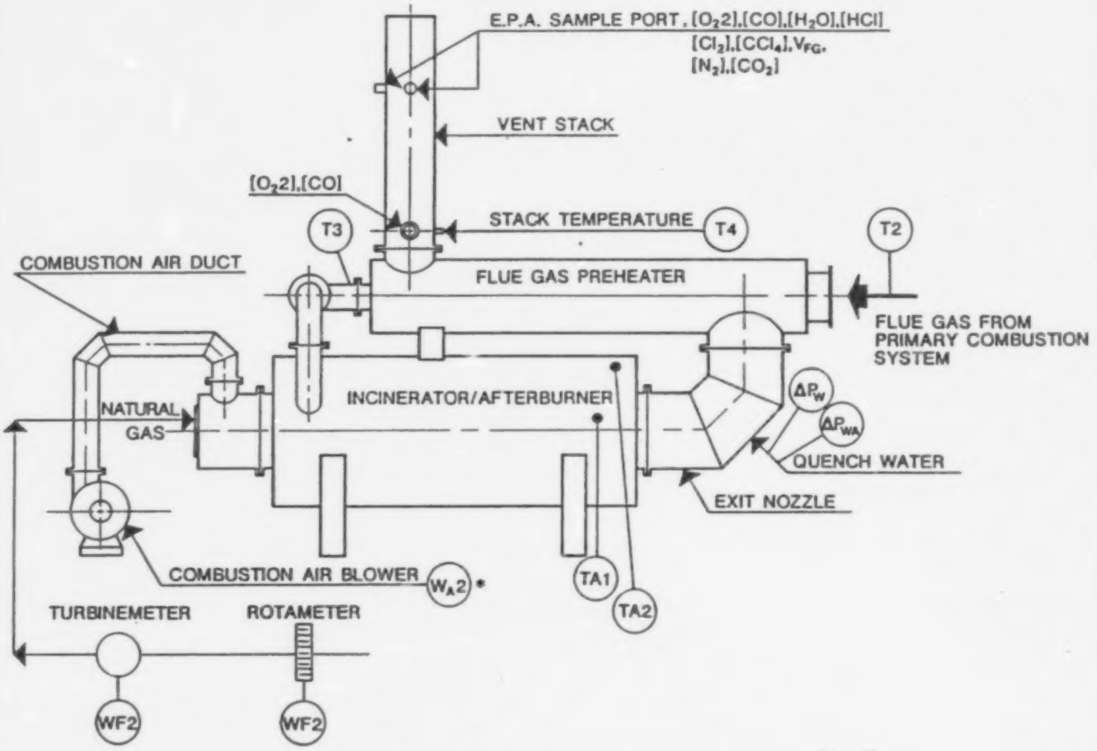

E.P.A. SAMPLE PORT, [$O_2$2],[CO],[H_2O],[HCl]
[Cl_2],[CCl_4],V_{FG},
[N_2],[CO_2]
VENT STACK
[$O_2$2],[CO]
STACK TEMPERATURE
T4
T3
T2
COMBUSTION AIR DUCT
FLUE GAS PREHEATER
FLUE GAS FROM PRIMARY COMBUSTION SYSTEM
NATURAL GAS
INCINERATOR/AFTERBURNER
ΔP_W
ΔP_{WA}
QUENCH WATER
EXIT NOZZLE
COMBUSTION AIR BLOWER
W_A2 *
TA1
TA2
TURBINEMETER
ROTAMETER
WF2
WF2

FIGURE 9 INCINERATOR/AFTERBURNER PARAMETERS

TABLE VIII - I/A MEASURED PARAMETERS AND INSTRUMENTATION

Parameter	Instrument
Natural gas flow rate*	Turbine model LO-II manufactured by Halliburton with a capacity of 10.0MM BtuH
Oxygen concentration*, (O_22)	Cell type model 326 AI-20 manufactured by Teledyne.
Flue gas inlet temp., T2	Type K thermocouple manufactured by Westronics.
Flue gas preheat temp., T3	Type K thermocouple manufactured by Westronics.
Operating temperature*, TA1	Type K thermocouple and indicator model 660 manufactured by Omega Engineering.
Operating temperature*, TA2	Type K thermocouple and indicator model 660 manufactured by Omega Engineering.
Vent stack temp., T4	Type K thermocouple and indicator model 660 manufactured by Westronics.
Carbon Monoxide*, CO	Infra Red Model-AR-500R manufactured by Anarad.
Data Logger	Hewlett Packard Model 3054DL with Hewlett Packard HP-85 Micro computer readout.

* continuously monitored

Figure 10 is a schematic of the hazardous waste feed system designed to supply a gaseous POHC at a constant and accurately measured flow rate to the flue gas interconnecting duct. Carbon tetrachloride is selected for the POHC because it is the fourth most difficult compound to incinerate.

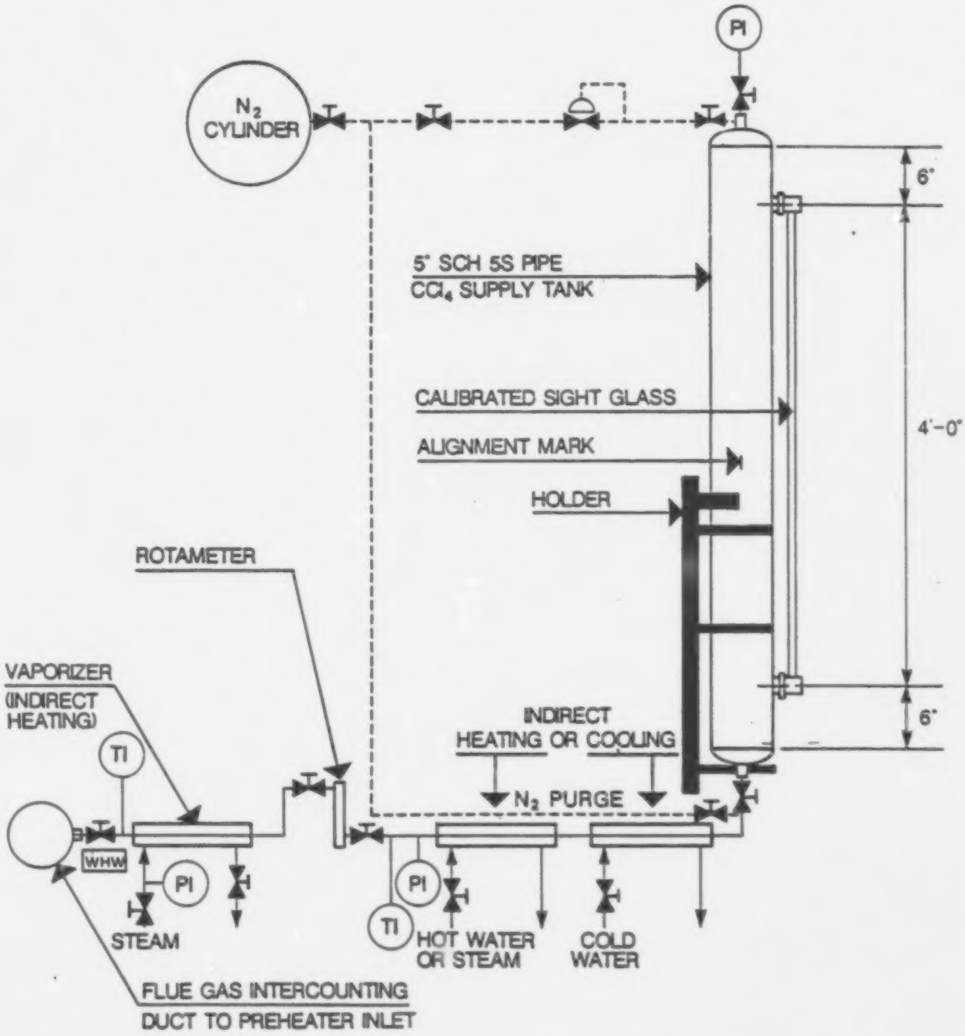

PI
N_2 CYLINDER
6"
5" SCH 5S PIPE
CCl_4 SUPPLY TANK
CALIBRATED SIGHT GLASS
ALIGNMENT MARK
HOLDER
4'-0"
ROTAMETER
VAPORIZER (INDIRECT HEATING)
TI
INDIRECT HEATING OR COOLING
6"
N_2 PURGE
WHW
PI
STEAM
PI
TI
HOT WATER OR STEAM
COLD WATER
FLUE GAS INTERCOUNTING DUCT TO PREHEATER INLET

FIGURE 10 HAZARDOUS WASTE FEED SYSTEM

The properties of CCl_4 follow:

Molecular weight, lbs/mole	153.84
Density, lbs/ft^3	99.52 (20°C, 68°F)
Boiling point, °F	170.2 (76.8°C)
Colorless liquid virtually insoluble in water.	

The equation describing the CCl_4 reaction with a stoichiometric amount of oxygen in the presence of water vapor is:

$$CCl_4^* + O_2 + 3.76N_2 + XH_2O \rightarrow CO_2 + 4HCl + (X-2)H_2O + 3.76N_2$$

Thus, four moles of hydrochloric acid (HCl) are produced for every mole of CCl_4 supplied to the I/A.

The allowable amount of HCl that can be emitted to the atmosphere is 1.8 kilograms per hour. This is equivalent to 3.965 lbs/hr which is 0.1087 lb moles/hr. Thus, 0.02718 lb moles/hr (0.1087/4) of CCl_4 is equivalent to a liquid rate of 19.82 ml/min.

The CCl_4 supply system consists of:

1. CCl_4 supply tank with a nominal size 5.5 inches x 5'-0" in length with an internal volume of approximately 0.78 ft^3. The volume included within the sight glass is approximately 0.57 ft^3.
2. N_2 cylinder for supply tank pressurization and line purge.
3. Indirect CCl_4 cooler using cold water for cooling.
4. Indirect CCl_4 heater using either hot water or steam for heating.
5. Indirect CCl_4 vaporizer using steam for heating.
6. Twelve (12) block valves
7. Nitrogen pressure regulator
8. Instrumentation

   Supply tank pressure indicator
   Rotameter inlet pressure indicator
   Rotameter inlet temperature indicator
   Vaporizer steam pressure indicator
   Vaporizer outlet temperature indicator

   a. Rotameter for indicator of CCl_4 flow rate.
   b. Sight glass for indicator of CCl_4 flow rate.

9. Lines (Tubing)

   Connecting N_2 cylinder to supply tank
   Connecting supply tank to indirect cooler
   Connecting indirect cooler to indirect heater

*Mathematically correct, but not intended as the only path to this result.

CALIBRATION PROCEDURES

Incinerator/Afterburner

A. Anarad Carbon Monoxide Meter

1. This CO meter is equipped with a zero and barometric calibration integral to the unit.
2. The instrument technician zeroes the meter and sets the barometric pressure. In addition, he spans the meter with a low (50 ppm) and a high (150 ppm) span gas that has been mixed and analyzed. These calibrations are performed a minimum of once daily during operation.

B. Teledyne Oxygen Meter

1. This oxygen meter has a high, medium, and low scale with ambient oxygen content marked on the scale integral to the unit.
2. The instrument technician zeroes the meter and then calibrates it against ambient oxygen. In addition, he spans the meter with two analyzed known mixtures consisting of a a low of approximately 0.5% 0_2 and a high of approximately 3.0%. These calibrations are performed a minimum of once daily during operation.

C. Thermocouples and Temperature Indicators

Thermocouples and temperature indicators are tested and calibrated using a Transmation PPS digital calibrator Model 1040. This calibration is performed a minimum of once a month.

D. Natural Gas Turbine Meter

This meter is calibrated against an in-line dry test meter. This calibration is performed a minimum of once a month.

E. Alarms and Automatic Shutdowns

Alarms and automatic shutdowns are tested weekly and prior to initial start-up of the units.

CCl_4 Feed System

The sight glass is calibrated as follows:

1. Position supply tank in holder and align tank mark with holder mark.
2. Bleed nitrogen through tank to remove any liquids or solids.
3. Close all supply tank valves.
4. Remove supply tank pressure indicator.
5. Attach funnel to supply tank inlet valve.
6. Open supply tank inlet valve.
7. Measure 50 ml of municipal water with a graduated cylinder.
8. Pour water from graduated cylinder into supply tank via funnel.
9. Record sight glass reading.
10. Repeat steps 8 and 9 until entire length of sight glass is calibrated.
11. Close supply tank inlet valve.
12. Remove funnel.
13. Install supply tank pressure indicator.

The Rotameter calibration is checked as follows:

1. Water remains in supply tank from sight glass calibration sequence.
2. Open nitrogen cylinder valve.
3. Open nitrogen supply line valve.
4. Open supply tank pressure indicator valve.
5. Open supply tank inlet valve slowly and in combination with nitrogen pressure regulator. Adjust supply tank pressure to 20 psig.
6. Open supply tank outlet valve.
7. Open rotameter outlet valve.
8. Open system outlet valve.
9. Partially open rotameter inlet valve.
10. Adjust rotameter outlet valve and inlet valve to achieve a flow rate of 20.0 ml/min.
11. Record the time required to remove several 100 ml portions from supply tank.
12. Record rotameter inlet temperature during step 11.
13. Repeat steps 10, 11 and 12 for a flow rate of 15 ml/min.
14. Repeat steps 10, 11 and 12 for a flow rate of 10 ml/min.

TEST OPERATING PROCEDURE

The PCS is gradually heated to an operating temperature of 1600°F. Then the auxiliary fuel is increased to 3.0 MM BtuH and the tempering air rate adjusted to maintain an operating temperature of 1600°F. The PCS is operated until steady state is achieved. Final adjustment of the PCS is made to produce the required flue gas rate.

A single center point I/A stack velocity reading is obtained at the EPA sample port (see Figure 9) by using a 304 S.S. type pitot tube Dwyer model number 160-24. Flue gas temperature is obtained by thermocouple T4. This information is used to determine the approximate amount of combustion products exiting the I/A. In the event this value is less than 333.3 actual cubic feet per second, the PCS fuel rate is increased. In the event this value is greater than 350 actual cubic feet per second, the PCS fuel rate is decreased.

This procedure is continued until the flue gas rate is indicated to be between 333.3 and 350 actual cubic feet per second. Then the trial burn configuration is operated for an hour and checked for steady state operation.

Carbon tetrachloride is supplied at a rate of 19.5 ml/min (1.6% less than allowable) to be interconnecting duct by the CCl_4 feed system. This small rate does not affect I/A operation which has been stabilized at steady state conditions.

Process data is obtained by John Zink Company staff every 15 minutes during the trial burn, except for the data continuously recorded (marked by asterisk in Table VIII). In addition, technicians monitor operation of the PCS and I/A on almost a continuous basis.

The proposed I/A operating and data sampling schedule follows:

Completion Time (Elapsed Hours)	Activity
0:00	1. Stabilize I/A operating temperature at 1800°F.
1:00	2. Introduce the surrogate POHC to the interconnecting duct.
2:00	3. Equilibrate the I/A on feed for 60 minutes.
3:00	4. Velocity, Traverse, Moisture, Hydrochloric Acid and Chlorine sampled from the same train. This train leak tested and sampling started simultaneously.
3:00	5. The unburned hydrocarbons and fixed gases will be collected during the above sampling time of 1 hour.
3:00	6. The VOST method assembled and leak tested at the same time the above train is tested. It is also sampled simultaneously with the above train in a different port. This method will require less than 20 minutes.
	7. The samples should be collected in 1 hour.
4:00	8. One hour will be required for dismantling, assembling, and leak testing for the second replicate run as above.
5:00	9. Completion of second replicate.
6:00	10. Repeat No. 8.
7:00	11. Completion of third replicate.
7:15	12. Discontinue surrogate POHC.
7:30	13. Dismantle equipment.
8:00	14. Follow-up burn to purge POHC @ 1800.
10:00	15. Normal shutdown and purge.
Summary:	POHC burn total 6 hr. 15 min. Total firing time 14 hrs. (estimated) Should any of the replicates be unacceptable due to sampling procedural requirements, the burn would be repeated the following day.

The proposed CCl_4 Feed System Operation follows:

1. Position supply tank in holder and align tank mark with holder mark.
2. Bleed nitrogen through tank to remove any liquids or solids.
3. Close all supply tank valves.
4. Remove supply tank pressure indicator.
5. Attach funnel to supply tank inlet valve.
6. Open supply tank inlet valve.
7. Fill supply tank with CCl_4.
8. Close supply tank inlet valve.
9. Remove funnel.
10. Remove supply tank with three valves and sight glass and weigh.
11. Position supply tank in holder and align tank.
12. Install supply tank pressure indicator.
13. Open nitrogen cylinder valve.
14. Open nitrogen supply line valve.
15. Open supply tank pressure indicator valve.
16. Open supply tank inlet valve slowly and in combination with nitrogen pressure regulator adjust supply tank pressure to 20 psig.
17. Open supply tank outlet valve.
18. Open rotameter outlet valve.
19. Open system outlet valve.
20. Partially open rotameter inlet valve.
21. Adjust rotameter outlet valve and inlet valve to achieve a flow rate of 15 ml./min.
22. Adjust rotameter inlet temperature to 68°F ± 2°F by adjusting either the cooler water inlet valve or heater water (or steam supply) inlet valve.
23. Adjust vaporizer outlet temperature indicator valve to 200°F or greater by adjusting the vaporizer steam inlet and outlet valves.
24. Adjust rotameter outlet valve and inlet valve to achieve a flow rate of 19.5 ml/min.
25. Repeat steps 22 and 23.
26. Close supply tank outlet valve at completion of analytical portion of trial burn.
27. Close nitrogen supply line valve.
28. Close supply tank inlet valve.
29. Open nitrogen purge valve and purge line for 5 minutes.
30. Close nitrogen purge valve.
31. Close nitrogen cylinder valve.
32. Close all valves.
33. Remove supply tank with three valves and sight glass and weigh.
34. Transfer any CCl_4 remaining in supply tank to original container.
35. Repeat steps 11, 12, 13, 14, 15, 16, 17 and 18.
36. Open rotameter inlet valve.
37. Open system outlet valve.
38. Operate system for 30 minutes.
39. Close all open valves.

The Sampling and Analytical Methods follow:

Parameter	Method
Velocity	CFR Part 40 Method 2
Traverse	CFR Part 40 Method 1
Moisture	CFR Part 40 Method 4
Unburned Hydrocarbons in Flue Gas	Integrated sampling procedures using gas sampling bags analyzed by GC - thermal conductivity. CFR Part 40 Method 3-B.
Fixed Gases in Flue Gases	Integrated sampling procedures using gas sampling bags analyzed by GC - thermal conductivity. CFR Part 40 Method 3-B.
Carbon tetrachloride	Protocol for the Collection and Analysis of volatile POHC's using "FAST-VOST" with three replicate tests for quality control.
Hydrochloric Acid	NIOSH Manual of Analytical Methods, Volume 3 S-246
Chlorine	Supelco Bulletin 769 Impinger Bubbler DOD Colormetric Procedure.

TRIAL BURN REPORT

EXCEPTIONS TO TRIAL BURN PLAN

All deviations from the Trial Burn Plan follow:

Indirect CCl_4 cooler - a double-pipe type heat exchanger using city water for cooling the CCl_4. This item was omitted because its function could be provided by the indirect CCl_4 heat exchanger.

CCl_4 supply tank support - the trial burn test plan proposed a tank support constructed so it could be separated from its stand. This proved to be impractical because of alignment problems. Welding the stand to the tank eliminated this problem.

A 15 psig supply tank pressure was used instead of 20 psig.

CCl_4 tank calibration was done with 1.0 liter samples instead of 100 ml samples.

CCl_4 was supplied at 19.8 ml/min vs. 19.5 ml/min.

PCS operated at 1640°F vs. the proposed 1600° F.

I/A operated at 1762 and 1513°F vs. the proposed 1800°F.

MEASURED DATA

The trial burns were conducted between 2:42 to 5:15 pm and 6:18 to 8:42 pm on April 12, 1985.

The PCS parameters measured by John Zink Engineers every 15 minutes from 2:10 pm until 8:55 pm are listed in Table IX.

TABLE IX PCS MEASURED PARAMETERS

Operating temperature, T1
Saturation temperature, TS
Natural gas flow rate, W_F1
Saturation pressure, PS (Venturi ΔP)
Barometric pressure, PB
Combustion air rate, $W_{CA}1$
Conditioning air rate, WQ
Flue gas oxygen concentration, (O_21)
Natural gas pressure, ΔP_{F1}
Quench pot level, HI
Separator level, H2

The I/A parameters measured by John Zink Engineers every 15 minutes from 2:10 pm until 8:55 pm are listed in Table X.

TABLE X - I/A MEASURED PARAMETERS

Operating temperature*, TA1 and TA2
Natural gas flow rate* (turbine), W_F2
Natural gas flow rate (rotameter), W_F2
Flue gas oxygen concentration*, (O_22)
Flue gas CO concentration*, (CO)
Flue gas inlet temperature, T2
Flue gas preheat temperature, T3
Vent stack temperature, T4
Combustion air rate, $W_{CA}2$

* Recorded every 2 minutes from 1:52 pm until 9:18 pm.

The CCl_4 flow rate was to be measured every 15 minutes but was measured every minute. This was required to improve accuracy because of the non-steady-state nature of the CCl_4 rotameter which required continual adjustment by a technician.

An analytical company measured the I/A flue gas concentration of: Oxygen (O_22); Carbon Monoxide (CO); Water (H_2O); Hydrochloric Acid (HCl); Chlorine (Cl_2); Carbon Tetrachloride (CCl_4); Nitrogen (N_2); and Carbon Dioxide (CO_2). They also measured Flue Gas Velocity, F_{FG}; Stack Temperature, T_4 and Barometric Pressure, P_B.

The analytical company obtained six sets of samples for determination of the emission rates of VOC(s), HCl and Cl_2. The first set of three runs, Test I, for an I/A operating temperature of approximately 1800°F were made from 2:42 to 3:14 pm, 3:45 to 4:15 pm and 4:48 to 5:15 pm. The second set of three runs, Test II, for an I/A operating temperature of approximately 1500°F was made from 6:18 to 6:46 pm, 7:12 to 7:42 pm and 8:16 to 8:42 pm. Immediately following each run two "grab" samples were obtained for determining the concentration of O_2, N_2, CO and CO_2. These "grab" samples were obtained at: 3:20 and 3:26 pm, 4:15 and 4:24 pm, 5:24 and 5:52 pm, and 8:48 and 8:55 pm. The concentration of water in the flue gas was determined during each run.

REDUCED DATA AND PROCEDURE

The data listed in Table XI represent average values. Each set of data is reduced to the degree which permits its use in energy and mass balance calculations.

Table XI - Analytical Laboratory Data, lb-mole/hr.

Test No.				I				II
Run No.	1	2	3		4	5	6	
Stack Flow	654.4	719.5	722.5	698.8	698.9	731.1	718.5	726.3
Component								
$HCl \times 10^2$	9.95	9.72	11.12	10.26	11.23	11.29	10.40	10.97
$Cl \times 10^5$	20.04	13.20	18.88	17.37	8.579	9.073	9.916	9.189
$CCl_4 \times 10^9$	8.541	12.44	29.74	16.91	23.07	16.44	3.965	14.49

Run No. 1 is used to explain how these values were determined. The stack velocity was reported as 78.667 ft/sec. Thus, the flue gas rate in lb-mole/hr, (M) was determined by the relationship:

$$M = (Vs \times 3600 \text{ sec/hr} \times As)/(V_R\ (TS/T_R)(P_R/P_B))$$

Where
V_R = 379.5 ft^3/lb-mole @ 520°R and 29.92 in. of Hg.
V_s = stack velocity ft/sec
A_s = stack flow area, $ft.^2$
T_R = 520°F (Temperature reference)
T_s = stack temperature, °R
P_B = barometric pressure, in. of Hg
P_R = 29.92 in of Hg (Pressure reference)

Thus, inserting values into the relationship gives:

$$\begin{aligned} M &= (78.667^* \ 3600^* \ 3.142)/(379.5(1825/520)(29.92/29.31) \\ &= 654.4 \text{ lb-mole/hr} \end{aligned}$$

which is the value shown in Table XI

The rates in lb-mole/hr for HCl, Cl, and CCl_4, were determined by first multiplying the reported mass concentration in lb/DSCF by the reported volumetric flow rate in DSCF/hr and then dividing by the molecular weight of the compound of interest. Thus, for HCl

$$M_{HCl} = 125118.7 \times 2.90 \times 10^{-5}/36.465 = 0.0995 \text{ lb-mole/hr}$$

which is the value shown in Table XI. The molecular weight used for HCl, Cl and CCl_4 are 36.465, 35.457 and 153.84 respectively.

These values are the results of Cl balance calculations. Table XII lists the amount of available chlorine as Cl and the equivalent amount of CCl_4 for Tests I and II.

Table XII - Chlorine as Cl and Equivalent CCl_4

	Rate, lb-mole/hr		
Component	Average	Equiv. Cl	Equiv. CCl_4
HCl	10.2×10^{-2}	10.26×10^{-2}	2.565×10^{-2}
Cl	17.37×10^{-5}	17.37×10^{-5}	4.342×10^{-5}
CCl_4	16.91×10^{-9}	67.64×10^{-9}	16.91×10^{-9}
	Sum Test I	0.1028	0.02569
HCl	10.97×10^{-2}	10.97×10^{-2}	2.742×10^{-2}
Cl	9.189×10^{-5}	9.189×10^{-5}	2.968×10^{-5}
CCl_4	14.49×10^{-9}	57.96×10^{-9}	14.49×10^{-9}
	Sum Test II	0.1098	0.02745

Table XIII lists the concentration of the fixed gases: O_2, N_2, CO and CO_2 on a wet basis. Included are all "grab" samples, two made after each run. Please note: the concentration of CO was reported to be less than 0.005 mole percent for all twelve grab samples so it was not listed in Table XIII.

Table XIII - Fixed Gases on a Wet Basis, Mole %

Run	1	1	2	2	3	3
Time	3:20 pm	3:26 pm	4:19 pm	4:24 pm	5:24 pm	5:32 pm
O_2	2.383	2.408	0.720	0.898	3.251	0.734
N_2	42.483	42.454	53.260	49.560	47.110	54.445
CO_2	4.784	4.788	.003	3.522	4.839	.021
H_2O	50.350	50.350	46.020	46.020	44.800	44.800
Run	4	4	5	5	6	6
Time	6:48 pm	6:53 pm	7:55 pm	7:58 pm	8:48 pm	8:55 pm
O_2	5.072	3.143	3.319	3.300	0.867	0.872
N_2	51.987	54.296	52.716	52.578	63.930	64.350
CO_2	4.571	4.191	3.885	4.042	0.473	0.048
H_2O	38.370	38.370	40.080	40.080	34.730	34.730

Table XIV is the average of the average process data recorded by the data logger every two minutes during Test I and II.

Table XIV - Average Data Logger Data

Run No.	W_F2, SCFH	(O_2), %	(CO),ppmv	TA1,°F	TA2,°F
			TEST I		
1	9904	3.72	36.6	1736	1729
2	9983	5.16	31.5	1811	1802
3	9951	4.92	21.7	1747	1742
Average	9946	4.60	29.9	1765	1758
			TEST II		
4	7888	7.29	14.05	1541	1531
5	7870	7.19	13.90	1477	1474
6	7858	7.36	14.7	1529	1522
Average	7872	7.28	14.22	1516	1510

Table XV is the average of the I/A process data recorded by John Zink engineers every 15 minutes during Tests I and II.

TABLE XV - Average John Zink Recorded I/A Data

						TEST I		
Run No.	TA1 °F	T4 °F	T3 °F	T2 °F	$W_f'2$ %	Nat. Gas ΔP Psi	WA2ΔP Psi	Quench Gun ΔP (psi) air/water
						TEST I		
1	1726	1348	333	189	72.5	30	1.69	92/74
2	1788	1355	332	192	73.0	30	2.00	95/75.3
3	1751	1306	322	184	73.0	30	2.00	86/74
Average	1755	1336	329	192	72.8	30	1.90	91/74.4
						TEST II		
4	1532	1179	298	186	60	30	2.00	87.3/70.7
5	1479	1160	296	189	60	30	2.00	87.5/71.5
6	1531	1174	296	183	60	30	2.00	86/70
Average	1514	1171	297	186	60	30	2.00	86.9/70.7

Table XVI lists the averages for the CCl_4 rotameter data recorded by John Zink Engineers every minute during Tests I and II.

Table XVI - Average CCl_4 Rotameter Flow Rate Data

Run No.	Rotameter, %
1	98.4
2	98.5
3	97.6
Test I Average	98.17%
4	96.5
5	96.8
6	96.7
Test II Average	96.67%

Table XVII lists the averages for the CCl_4 tank sight glass data recorded by John Zink engineers every 15 minutes during Tests I and II.

Table XVII - Average CCl_4 Tank Sight Glass Flow Rate Data

Run No.	Time		ΔT,Min	Height		ΔH,in	ΔH/ΔT,in/Min
	Start	End		Start	End		
1	2:45	3:16	31	48.75	47.50	1.25	0.0403
2	3:31	4:15	44	46.75	44.75	2.00	0.0454
3	4:51	5:17	26	41.78	43.00	1.125	0.0433
				Test I Average			0.043 in/min
4	6:47	6:15	32	40.312	39.00	1.312	0.0410
5	7:55	7:03	52	38.25	36.00	2.25	0.0433
6	8:48	8:12	36	35.375	33.75	1.625	0.0451
				Test II Average			0.0431 in/min

Table XVIII lists the average of the process data recorded by John Zink engineers every 15 minutes. The Test I and II values are averages of the average data.

Table XVIII-PCS Process Data

				TEST I				
Run No.	T1	TS	($O_2$1)	W_{CA}1,ΔP	PS	W_F1,ΔP	ΔP_F1	WQ,ΔP
	(°F)	(°F)	(%)	(*)	(*)	(%)	(PSI)	(*)
1	1635	183	13.1	7.05	12.8	94	31	41.5
2	1641	190	12.8	7.07	14.3	94	31	41.8
3	1643	187	13.1	7.00	14.0	94	31	41.5
Average	1640	187	13	7.04	13.7	94	31	41.6
				TEST II				
4	1635	179	13.2	7.17	11.3	94	31	41.8
5	1641	181	12.8	7.10	12.0	94	31	41.5
6	1643	181	12.6	7.20	11.0	94	31	42.0
Average	1640	180	12.9	7.16	11.4	94	31	41.8

* - in. of w.c.

Data Evaluation

The data obtained by the Analytical Laboratory was evaluated with respect to consistency and accuracy where appropriate.

The stack flow averages listed in Table XI seem reasonable, even though the stack flow tends to increase during the course of the runs.

The HCl, Cl, and CCl_4 averages listed in Table XI seem reasonable with the exception of the CCl_4 average for Test II. The Run 4 CCl_4 flow rate is 23.07×10^{-9} lb-mole/hr, Run 5; 16.44×10^{-9} lb-mole/hr and Run 6; 3.965×10^{-9} lb-mole/hr. The 3.965×10^{-9} value is inconsistent with Run 4 and 5 values. Eliminating Run 6 from the CCl_4 average would increase the average value for Test II from 14.49×10^{-9} to 19.76×10^{-9} lb-mole/hr. This would be a more reasonable value when calculating DRE. The values listed in Table XII seem reasonable, except Test II requires modification if the CCl_4 flow rate of 3.965×10^{-9} lb-mole/hr is questioned. Making this correction modifies the CCl_4 line data as follows:

19.76×10^{-9} instead of 14.49×10^{-9}
79.04×10^{-9} instead of 57.96×10^{-9}
19.76×10^{-9} instead of 14.49×10^{-9}

This correction does not change the Test II value because of the insignificance of the addition of 5.27×10^{-9} lb-mole/hr of CCl_4 to the averages.

Several values listed in Table XIII for the concentration of the fixed gases in the flue gas for both Test I and II seem to be inaccurate. The concentration of O_2 and CO_2 for Run No. 2 at 4:19 pm, the O_2 for Run No. 2 at 4:24 pm, the CO_2 and O_2 for Run No. 3 at 5:32 pm, the O_2 and CO_2 for Run No. 6 at 8:48 pm, and the O_2 and CO_2 for Run No. 6 at 8:55 appear to be incorrect when compared to the steady-state operation of both the PCS and I/A. Eliminating these runs from consideration and averaging the remaining gives the information listed in Table XIX.

Table-XIX Fixed Gas on a Wet Basis (Modified)

Run	1	1	3		Test I
Time	3:20 pm	3:26 pm	5:24 pm		
Component, Mole%					
O_2	2.383	2.408	3.251		2.681
N_2	42.483	42.454	47.110		44.015
CO_2	4.784	4.788	4.839		4.804
H_2O	50.350	50.350	44.800		48.500
					100.0%

Run	4	4	5	5	Test II
Time	6:48 pm	6:53 pm	7:55 pm	7:58 pm	
Component, Mole %					
O_2	5.072	3.143	3.319	3.300	3.708
N_2	51.987	54.296	52.716	52.578	52.895
CO_2	4.571	4.191	3.885	4.042	4.172
H_2O	38.370	38.370	40.080	40.080	39.225
					100.0%

The average data logger data listed in Table XIV appears to be very consistent. The natural gas flow rate during Test I is higher than Test II. The concentration of O_2 in Test II is higher than Test I reflecting the decrease in O_2 consumption by a less natural gas flow rate while all combustion and quench air rates were essentially constant. Both Afterburner operating temperatures decreased with the decrease in the natural gas flow rate as expected.

The decrease in the concentration of CO in Test II operating at a lower temperature versus the higher operating temperature Test I was unexpected. It appears that the increase in the concentration of O_2 had more influence than the decrease in temperature, which is not the historical rule for an O_2 concentration exceeding 3%.

All the average data listed in Table XV for the I/A appears very consistent. The only data appearing to be out of order is the combustion air flow, WA2 for Run 1.

The I/A fuel firing rate was measured by a Halliburton turbine meter and a Brooks Rotameter. Table XIV reports values for W_F2 of 9,946 and 7,872 SCFH for Tests I and II respectively. Table XV reports valves for $W_F'2$ of 72.8 and 60% for Tests I and II respectively.

The values for W_F'2 are converted into MM BtuH by multiplying them by 950 Btu/SCF. The values for W_F2 are converted into MM BtuH by using rotameter flow correlation equations. Performing this conversion gives the values listed in Table XX.

Table XX - I/A Fuel Firing Rate

Test No.	W_F'2, MM BtuH	W_F'2, MM BtuH
I	9.449	8.743
II	7.478	7.206

The fuel rate measured by the turbine meter gives a higher fuel firing rate than the rotameter. The Test I rate is approximately 7.5% higher and the Test II rate is approximately 3.5% higher which averages to approximately 5.5%. This difference is within normal rotameter tolerance.

The average CCl_4 flow values measured by the Rotameter listed in Table XVI for each run are shown to be very consistent.

The average CCl_4 values measured by the CCl_4 tank sight glass listed in Table XVII for each run are shown to be very consistent.

All the average values listed in Table XVIII for the PCS for each run are shown to be very consistent. It operated as close as practical to steady-state conditions.

PERFORMANCE ANALYSIS

CCl_4 Destruction Removal Efficiency

The Destruction Removal Efficiency, DRE is determined by the relationship:

$$DRE = \frac{Win - Wout}{Win} \times 100\%$$

Where: Win = mass flow of a POHC in the waste
Wout = mass flow of a POHC entering atmosphere

The POHC of interest is CCl_4.

The amount of CCl_4 supplied to the I/A was measured directly by a rotameter and CCl_4 tank sight glass, and indirectly during the stack sampling by measuring the rate of compounds containing chlorine being discharged to the atmosphere.

The equivalent CCl_4 flow rate for Test I and Test II is 0.02569 and 0.02745 lb-mole/hr respectively (Table XII). This equates to a mass rate for Test I and Test II of 3.952 and 4.223 lb/hr respectively. The average values listed in Table XVI for Test I and Test II are 98.17 and 96.67% respectively, which are Rotameter readings. Use of the Rotameter flow correlation equation converts these readings into lb/hr. The mass rates for Test I and Test II are 4.142 and 4.079 lbs/hr respectively.

The average values for the rate of decrease of the CCl_4 tank liquid level are listed in Table XVII. These rates for Test I and Test II are 0.0430 and 0.0431 in./min. respectively. These values are converted into mass, W_{sg} (lb/hr) rate by the relationship.

$$
\begin{aligned}
W_{sg} &= Ao^{*}\,\Delta H/\Delta T\text{ (60 min/hr) DCC1 (1 ft/12 in)}\\
\text{Where Ao} &= \text{Combined tank/sight glass area}\\
&= 0.2038\text{ ft.}^2\,(0.0014 + 0.2024)\\
DCCL_4 &= CCl_4\text{ density }(62.4 \times 1.594 = 99.45\text{ lb/ft}^3)\\
\Delta H/\Delta T &= \text{Change in }CCl_4\text{ level, in./min.}
\end{aligned}
$$

This relationship determines for Test I and Test II a CCl_4 flow rate of 4.358 and 4.368 lb/hr respectively.

Table XXI makes a comparison of the CCl_4 flow rate determined by the three measurement methods.

Table XXI - CCl_4 Flow Rate (Comparison), lb./hr.

Test	Analytical	Rotameter	Sight Glass
I	3.952	4.142	4.358
II	4.223	4.079	4.368

The Analytical Laboratory and rotameter values for Test I and Test II vary by +4.8 and −3.4% which is good agreement when conducting a lab-type test on commercial size equipment.

The CCl_4 emission rate listed in Table XI for Test I and II is 16.91 and 14.49 $\times$ 10^{-9} lb-mole/hr respectively. A higher, more reasonable value instead of 14.49 $\times$ 10^{-9} lb-mole/hr which was discussed previously. Multiplying these values by the molecular weight of CCl_4 determines a mass rate for Test I and Test II of 2.601 and 3.040 x 10^{-6} lb/hr.

Substituting the CCl_4 emission rate of 3.040 x 10^{-6} lb/hr and the Analytical Company's CCl_4 rate of 4.223 lb/hr listed in Table XXI for Test II into the DRE relationship gives:

$$
\begin{aligned}
DRE &= \frac{(4.223\text{ lb/hr} - 3.040 \times 10^{-6}\text{ lb/hr})}{4.223\text{ lb/hr.}} \times 100\%\\
&= 99.99993\%
\end{aligned}
$$

Continuing with the substitution of the remaining CCl_4 emission rates and CCL_4 feed rates into the DRE relationship gives the calculated DRE values listed in Table XXII.

TABLE XXII-CCl_4 DRE, %

Test	Analytical	Rotameter	Sight Glass
I	99.9999	99.9999	99.9999
II	99.9999	99.9999	99.9999

HCl Emission Rate

The system was not fitted with an HCl scrubber because the POHC feed rate was controlled to meet the allowable HCl emission rate of 4.0 lb/hr or less. For Test I and Test II, Table XI listed an average HCl flow rate of 10.26 and 10.97 x 10^{-2} lb-mole/hr respectively. Multiplying this rate by the molecular weight of HCl gives, for Test 1 and Test 2, a mass rate of 3.741 and 4.000 lbs/hr respectively.

I/A Residence Time

I/A residence time is calculated by dividing the I/A combustion volume in the units cubic feet by the flue gas flow rate in the unit actual cubic feet per second (ACFS).

Table XI gives for Test I and Test II a stack flow rate of 698.8 and 716.2 lb-mole/hr respectively. The amount of water added at the exit nozzle for cooling the I/A flue gas must be subtracted from the stack flow rates to determine a "true" combustion chamber flow rate. The water rate is determined by using the values for the average quench gun atomizing air pressure and average water supply pressure and the correlation curve for the Delevan model 32740 air atomized water nozzle. The values of the set: (air ΔP, water ΔP) for Test I and Test II are (91.0,74.4) and (86.9,70.7) respectively.

Use of the correlation curve determined the water volumetric flow rates for Test I and Test II to be 1.24 and 1.26 gpm respectively. The 1.24 gpm is equivalent to 34.4 lb-mole/hr. and the 1.26 gpm flow rate is equivalent to 35.0 lb-mole/hr. Subtracting these values from the stack flow rate gives an I/A flow rate for Test I and Test II of 664.4 and 681.2 lb-mole/hr. respectively.

The flow of combustion products in the vent stack, (V) in actual cubic feet/sec (ACFS), is determined by the relationship.

$$V = M V_R (TO/T_R)(P_R/P_B)/3600 \text{ sec/hr}$$

where

M = Stack gas flow, lb-mole/hr
V_R = 379.5 ft^3/lb-mole @ 520°R and 29.92 in of Hg
T_O = Average Operating temperature, °R
T_R = 520°R (Temperature reference)
P_R = 29.92 in of Hg. (pressure reference)
P_B = Barometric pressure, in. of Hg.

For Test I (See Table XIV)

$$T_O = 2222\ °R\ ((1765 + 1758)/2 + 460)$$

For Test 2 (See Table XIV)

$$T_O = 1973\ °R\ ((1516 + 1510)/2 + 460)$$

The barometric pressure for Test I is 29.3 in. of Hg. and for Test II is 29.35 in. of Hg. Substituting these values into the relationship, determines the combustion product rates for Tests I and II to be 305.6 and 277.8 ACFS respectively. Thus, the gas residence time for each test is:

Test I: R_T = 1.09 seconds
Test II: R_T = 1.20 seconds

DRE Correlation

The capability of a time-temperature environment to destroy a hazardous waste is predicted by the first order decomposition/oxidation equation:

$$\text{DRE} = 100\,(1\text{-}C/Co) = 100\,(1 - E^{Kt}) \qquad (1)$$

Where
DRE = Destruction efficiency, %
C = Concentration at Time t
Co = Initial concentration
K = Arrhenius equation reaction rate, sec.^{-1}
t = Time, sec.

The Arrhenius equation for determination of the reaction rate is:

$$K = Ae^{-(E/RT)} \qquad (2)$$

where
A = Constant
E = Energy of activation, Btu/lb-mole
R = Universal gas constant, Btu/(lb-mole-°R)
T = Temperature, °R

Taking the natural log of equation (1) gives:

$$\text{Ln } K = A'\,(1/T) + B' \qquad (3)$$

Where A', B' = Constants

Equation (3) requires the performance points: $(K_1, I/T_1)$ and $(K_2, I/T_2)$ to solve for the constants A' and B'. The information listed in Table XXIII was determined previously.

Table XXIII - DRE Correlation Data

TEST	R_T, SEC	T, °R	DRE, %
I	1.09	2222 (1762 °F)	99.9999+
II	1.20	1973 (1513 °F)	99.9999+

Substitution of the DRE and R_T values into equation (1) and solving for K gives a value of 12.67 for Test I and 11.51 for Test 2. Substituting these K values and respective temperatures, into equation (3) determines values for the constants A' and B' to be –1690 and 3.2998 respectively. Equation (3) becomes:

$$\text{Ln } K = (-)1690\,(1/T) + 3.2998 \qquad (4)$$

With a temperature range applicability of 1500°F (1513-13) to 2000°F (1762 + 138°F).

I/A Capacity

The I/A is required to achieve a DRE of at least 99.99%. The required operating parameter couple of R_T and T can be determined by using equations (1) and (4). K values for: 1500, 1600, 1700, 1800, 1900, and 2000°F, as determined by equation 4, are 11.44, 11.93, 12.40, 12.83, 13.25 and 13.63 respectively. Substituting these K values and a DRE = 99.99% into equation (1) enables calculation of the required R_T. These R_T values for 1500, 1600, 1700, 1800, 1900 and 2000°F are 0.805, 0.722., 0.743, 0.718, 0.695 and 0.676 seconds respectively.

The flue gas rate in the I/A, VA in ACFS corresponding to these residence times is determined by dividing the I/A volume of 359 ft^3 by R_T. These VA values for: 1500, 1600, 1700, 1800, 1900, and 2000°F are 414, 431.71, 448.6, 464.2, 479.6, and 493 ACFS respectively.

Converting an I/A flue gas rate into a total I/A firing rate is accomplished as follows:

The reaction of CH_4 with a stoichiometric amount of O_2 is described by:

$$CH_4 + 20_2 = CO_2 + 2H$$ [illegible]

The stoichiometric air requirement, (MA), per lb-mole of CH_4 is:

$$MA = 9.52 \qquad (2/.21)$$

The 25% excess air requirement (MA25), per lb-mole of CH_4 is:

$$MA25 = 11.9 \qquad (9.52^{*}\ 1.25)$$

The resulting combustion products with 25% excess air are listed in Table XXIV.

Table XXIV - Fuel Combustion Products for 25% Excess Air

Component	Lb-Mole/Lb-Mole of CH_4
CO_2	1.0
H_2O	2.0
O_2	0.5 [(11.9-9.52) .21]
N_2	9.4 (11.9 x .79)
Sum of Products	12.9 lb-mole/lb-mole of CH_4

CH_4's lower heating value Lhv^T as a function of temperature given in reference tables is repeated in Table XXV.

Table XXV - Lower Heating Value for CH_4

Temperature (T), °F	Lhv^T, Btu/lb-mole
1500	193,100
1600	181,800
1700	170,300
1800	158,800
1900	147,100
2000	135,400

Use of quench air to control I/A operating temperature is a "worst case" from a residence time viewpoint, than the flue gas produced by a PCS because it does not contain water. Thus, a less volume of combustion products are produced by a particular I/A fuel firing rate.

The sensible heat of air, HS^T as a function of temperature given in the reference tables is repeated in Table XVI.

Table XXVI - Sensible Heat of Air

Temperature (T), °F	HS^T, Btu/lb-mole
1500	10,710
1600	11,520
1700	12,320
1800	13,140
1900	13,960
2000	14,790

The heat lost from the I/A by radiation and convection, Q_L to the atmosphere is 0.215×10^6 BtuH.

The heat balance for the system at temperature T is:

$$LHV^T (M_{CH4}) = Q_L + HS^T (M_{QA}) \qquad (5)$$

where M_{CH4} = CH_4 rate, lb-mole/nr

M_{QA} = quench air rate, lb-mole/hr

The flow gas product rate, M_{FG} (lb-mole/hr) is for a fuel rate of M_{CH4} and a quench air rate of M_{QA} is given by:

$$M_{FG} = M_{QA} + M_{CH4} (12.9) \qquad (6)$$

The volumetric flow rate of the combustion products, V_{FG} is given by:

$$V_{FG} = M_{FG} \frac{(379.5 \text{ ft}^3\text{/lb-mole})}{(3600 \text{ sec/hr})} \frac{(T+460°R)}{(520°R)} \frac{29.92 \text{ in. of Hg}}{(P_B \text{ in. of Hg.})}$$

$$= M_{FG} (6.066 \times 10^{-3}) (T+460)/P_B \qquad (7)$$

The allowable V_{FG} corresponding to a residence time is determined by:

$$V_{FG} = \frac{333.3}{R_T} \qquad (8)$$

Substituting the relationship for V_{FG} given by equation (8) into equation (7) gives:

$$M_{FG} (6.066 \times 10^{-3}) (T+460)/P_B = \frac{333.3}{R_T} \qquad (9)$$

Rearranging equation (9) gives:

$$M_{FG} = \frac{54946 P_B}{(T+460)R_T} \text{ lb-mole/hr} \qquad (10)$$

Substituting the values for T, P_B and R_T into equation (10) gives the information listed in Table XXVII.

Table XXVII - Allowable Flue Gas Flow Rate

Temperature, °F	M_{FG}, lb-mole/hr
1500	1020
1600	1012
1700	1003
1800	992.1
1900	981.5
2000	968.0

Substituting $Q = 0.215 \times 10^6$ into the energy equation (5) gives:

$$LHV^T (M_{CH4}) = 0.215 \times 10^6 + HS^T (M_{QA}) \qquad (11)$$

Equation (6) and (11) contain the known parameters: Lhv^T, HS^T and M_{FG}, at T, giving two equations with the two unknowns: M_{CH4} and M_{QA}.

At 1500°F these two equations become:

$$0.193 \times 10 (M_{CH4}) = 0.215 \times 10^6 + 10{,}710 (M_{QA}) \qquad (12)$$
$$\text{and } 1020 = M_{QA} + M_{CH4}\ 12.9$$

Solving each equation for M_{QA} gives

$$M_{QA} = 18.020\ M_{CH4} - 20.075 \qquad (13)$$
$$\text{and } M_{QA} = -12.9\ M_{CH4} + 1020 \qquad (14)$$

Subtracting equation (14) from equation (13) gives

$$0 = 30.92\ M_{CH4} - 1040.1$$

Solving for M_{CH4} gives

$$M_{CH4} = 33.64 \text{ lb-mole/hr.}$$

The CH_4 fuel firing mole rate is equivalent to a heat release at 60°F, Q60 of 11.484×10^6 Btu/hr ($33.64 \times 16.04 \times 21284$). This rate is calculated by multiplying the Methane rate; 33.6 lb-mole/hr by the molecular weight of Methane; 16.04 by the lower heating value of Methane at 60°F; 21284 BtuH.

Performing the same calculation for the other operating temperatures of interest determines the allowable firing rate as a function of temperature, these values are listed in Table XXVIII.

Table XXVIII - Allowable I/A Firing Rate

Temperature, °F	Q,60, BtuH
1500	11.484×10^6
1600	12.199×10^6
1700	13.648×10^6
1800	13.778×10^6
1900	14.523×10^6
2000	15.708×10^6

Assuming a 10% safety factor for uncertainty reduces the allowable fuel firing rate by 10% and increases available residence time by 10%.

Figure 11 is a plot of I/A firing rate and residence time versus I/A operating temperature required to achieve a DRE = 99.99%. The solid lines are for DRE test data and the dashed lines reflect the use of a 10% safety factor. The horizontal line represents the maximum burner firing capacity of 10.0MM BtuH.

According to Figure 11, the I/A will achieve a DRE of at least 99.99% when operated at a maximum fuel rate of 10.0 MM BtuH or less at any operating temperature over 1500°F.

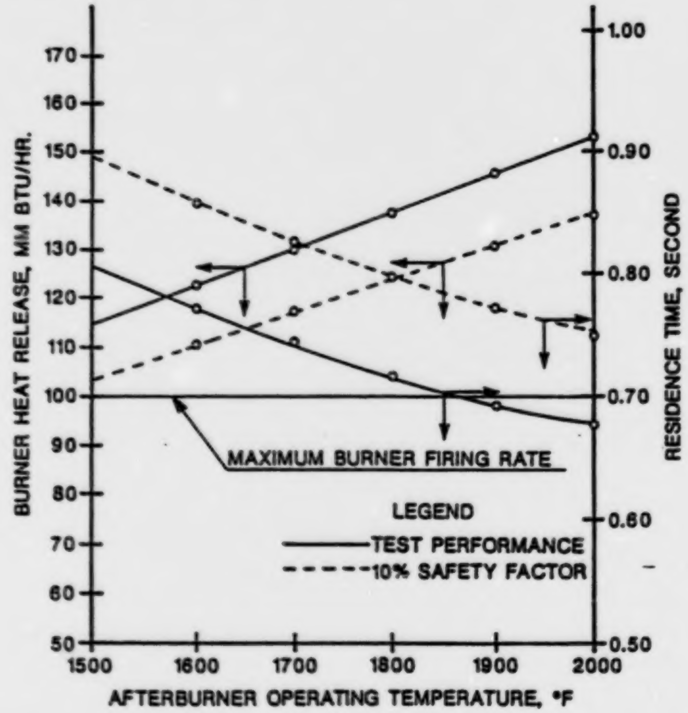


FIGURE 11 I/A OPERATION FOR A DRE=99.99%

PERFORMANCE SUMMARY

Each trial burn test achieved a DRE for CCl_4 in excess of 99.9999%. The values for the prime operating parameters are listed in Table XXIX.

Table XXIX - Prime Operating Parameters

Test Parameter	I	II
PCS firing rate, MM BtuH	3.196	3.196
I/A firing rate, MM BtuH	9.449	7.478
Test System firing rate, MM BtuH	12.647	10.674
I/A operating temperature, °F	1762	1513
I/A flue gas O_2, % Wet	2.68	3.71
% Dry	5.20	6.10

CONCLUSIONS

The PCS operated in a nearly steady-state fashion throughout the Trial Burn tests.

The Hazardous Waste Feed System operated in a nearly steady-state fashion throughout the Trial Burn tests.

The I/A operated in a nearly steady-state fashion at two test temperatures.

Data measured by Analytical Laboratory and John Zink was accurate and consistent.

The two I/A test temperatures achieved a DRE for CCl_4 in excess of 99.9999%.

The I/A can achieve a DRE for CCl_4 in excess of 99.99% when it operates at a temperature of 1500°F or greater and an auxiliary fuel firing rate is 10.0 MM BtuH or less.

The HCl produced during trial burn tests was equal to or less than 4.0 lbs/hr.

SECTION

XVII

HIGH TEMPERATURE LUMBER DRYING

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Graham Mackay
Forintek Canada Canada Corporation

9/5/666

ABOUT THE SPEAKER:

Graham Mackay obtained his B.Sc. and Ph.D. degrees from the University of Aberdeen, Scotland. He began his research career in lumber drying at the Forest Products Laboratory in Melbourne, Australia in 1968 where his activities included developing a new high temperature drying process for radiata pine. He came to the Western Forest Products Laboratory, now Forintek Canada Corp., in Vancouver in 1973, and is responsible for a program of basic and applied research in drying, and technology transfer to the lumber industry. Current areas of study include development of high-speed drying schedules for coastal hem-fir lumber.

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HIGH TEMPERATURE LUMBER DRYING

J.F. Graham Mackay
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The kiln drying of lumber in British Columbia and Western Canada in general can be categorized in several ways. There has always been a marked difference between the interior regions where the predominant species group is a mixture of white spruce, lodgepole pine and alpine fir (S-P-F), and the coastal regions where the major species are western red cedar and the species group of western hemlock and amabilis fir (hem-fir). Almost all of the wood cut in the interior is produced for framing and construction grades e.g. studs and joists, and is normally kiln dried. On the coast there has always been an effort to select the upper grades of lumber which are marketed and utilized as clears, that is lumber largely free of manufacturing and natural defects, and shop grades which upon further remanufacturing will yield high value products. This lumber too is normally kiln dried.

The balance of lumber produced on the coast has generally not been kiln dried, but has been exported green. To protect this lumber from mold and stain while in transit and storage surface chemical treatments have been applied. Increasing resistance to the use of these chemicals, and a recognition that a dried product is also a value added product, is promoting a move towards more kiln drying on the coast .

The difference in emphasis on kiln drying in the interior and on the coast came about largely due to transportation costs. From the interior, lumber is moved by road and rail and is charged according to weight. On the coast there is no such consideration since shipboard lumber is charged on a volume rather than weight basis.

A further distinction in kiln drying that can be made is in terms of the temperatures that are used; these temperatures relate closely to the characteristics of the wood and products being dried, and the required quality of the dried products.

The lowest temperature kilns operate at temperatures up to about 60°C. Typical examples are the electric heat pump dehumidifiers that are becoming widely used in the coastal region. These low temperature kilns are generally ideally suited to plants where volume throughput is not the major concern but where high quality is very important. Typical products are door and window joinery stock.

Medium temperature kilns, up to about 80°C, often use hot water or low pressure steam as the heat transfer medium. These kilns also are used for drying the upper grades of coastal lumber.

Kilns operating up to about 100°C typically are used in the interior for drying S.P.F. framing and construction grades.

High temperature drying is a loosely defined term describing drying at temperatures in excess of 100°C. Temperatures up to 120°C are used in B.C. for kiln drying the narrower widths of S-P-F, particularly studs. Traditionally these kilns have been natural gas-fired systems where the products of combustion of the gas are mixed with kiln air to develop a blend at about 175°C which is fed back to the kiln. An alternative heating source that competes with gas furnaces is a wood waste-fired combustor with thermal oil-filled coils to transfer heat to the kilns.

A fundamental difference exists between the mechanisms of moisture flow in wood during high temperature drying (above 100°C) and conventional drying (below 100°C). In conventional drying moisture movement from the wet core to the surface of the wood is by diffusion where the driving force is the partial vapour pressure gradient between the wet wood and the air. In high temperature drying a slight steam pressure is created within the wood resulting in mass flow of water vapour from the wet core to the wood surface. The ease with which the evaporated vapour can move from the inside to the outside of the wood largely determines whether high temperature drying can be used. If the vapour cannot readily escape, the internal pressures developed may be great enough to split the wood.

The following are some of the advantages of high temperature kiln drying:

The higher the air temperature the faster wood will dry, provided that neither visual nor structural damage to the products results. It is therefore possible to carry a lower inventory of lumber to produce a given volume of dried products. Similarly, less kiln capacity is required to produce a given annual throughput.

Less thermal energy is required for drying a given volume of wood. Because of the higher temperature of air in the kiln, the air "holds" more water vapour and thus less venting of humid air is required. Since drying times are shorter there is less loss of heat through the kiln structure.

High temperature kiln drying has been shown to be effective in reducing warp in warp-prone wood, for example 2 x 4's sawn from small pine logs often twist during conventional drying but stay flat during high temperature drying. This lumber is top-loaded with concrete blocks in the kiln and dried at 120°C. The high temperature plasticizes the wood while the weights hold it flat. When allowed to cool after drying, still under restraint, the lumber remains flat unless it is rewetted. This technique has had significant impact in the conversion of small softwood thinnings from the massive plantations of Australia and New Zealand, and also in the conversion of small southern pine logs in the United States.

There are also some well recognized disadvantages of high temperature kiln drying;

In all forms of kiln drying, thee layers of the wood dry and shrink before the inner layers do so. Shrinkage stresses develop on the surface of the wood and if these exceed the strength of the wood, checks and splits result .

High temperature drying normally causes very high moisture gradients and therefore some degree of surface checking, which while not impacting on wood strength, is not acceptable for clear and shop grades of lumber. For this reason high temperature drying should only be used for non-appearance grades.

The surfaces of some woods darken appreciably in colour when high temperature dried, thus if appearance is a consideration and the wood is not to be planed, high temperatures should not be used.

Moisture content variation both within and between boards tends to be greater in some species when high temperature dried.

A more important consideration is whether high temperature drying causes structural or mechanical damage to the wood, thereby lessening its strength or engineering properties. This subject has been exhaustively researched and for western softwood species there is evidence that in some circumstances high temperature drying can result in reduced bending strength although not in bending stiffness. Three interacting factors contribute to strength loss, moisture content, temperature and time of exposure. It is generally agreed that the hotter and wetter that wood is, and the longer it is heated, the greater will be the reduction in strength. These factors must be addressed in the design of a high temperature kiln.

Modern conventional kilns are built with an aluminum or steel frame on which are placed prefabricated insulated aluminum panels. Other than consider thicker panels to enclose more insulation and provide greater heat efficiency this structure is adequate for high temperature drying.

Air circulation systems are presently designed to provide a flow of 3-4 m/s through lumber rows separated by 19mm spacers. In high temperature drying rapid mass flow of moisture to surfaces occur, therefore use can be made of higher air flow, at least 6 m/s, to remove the moisture and provide heat to the wood, and therefore to minimize drying time. A recently completed study of high temperature drying of southern pine demonstrated reductions in drying times of about 25 percent by increasing air flow rate from 6 to 10 m/s.

To accomplish the high air flow high capacity, cross-mounted fans are required. Unlike conventional kilns where motors are direct-mounted to fans, motors should be mounted outside of a high temperature kiln with a short shaft to drive each fan.

To heat a high temperature kiln either one of the following can or is being used; for indirect heated kilns wood waste-fired combustors with high pressure steam or thermal oil, and for direct heated kilns natural gas-fired hot air. The distribution of heat is an important consideration in order to achieve minimum time of exposure to high Temperatures. For example a good high temperature kiln should have sufficient heat capacity to reach 120°C in 1 hour or 150°C in 2 hours.

The placement of lumber and the location of heat exchangers in high temperature kilns should differ significantly from that in conventional kilns. In a typical double track kiln with a 3 meter-wide load on each track, air must pass through 6m of lumber before being reheated. This results in lumber towards the outer sides of the kiln drying faster than lumber located towards the centre of the kiln. The 'outer' lumber is dry and ready to exit the kiln part way through the drying cycle but has to remain in the kiln until the 'inner' lumber is dried to the desired moisture content. Because conventional temperatures, unlike high temperatures, do not cause strength loss in wood, the extended exposure does not pose a problem.

An ideal high temperature drying kiln is a narrow structure, wide enough to accommodate only a 1.5 m-wide load so that rapid drying proceeds uniformly without part of the load being overdried or needlessly exposed to damaging conditions. Alternatively, a conventional double-track kiln would convert to a 4-track kiln, each track 1.5 m wide with a heat exchanger between each of the single tracks so that the target dry bulb temperature can be re-established at the air entry side of each track. Thus all the advantages of high temperature drying would be accomplished without inducing strength loss.

The potential for greater or new use of high temperature kiln drying in British Columbia is considerable. On the coast there is an opportunity to significantly reduce drying times for construction grades of hem-fir. Long drying times are required for these products in conventional temperature kilns, contributing to high drying costs and a virtual abandonment of the practice. High temperature drying could reverse this trend. In the interior a limited amount of high temperature drying is presently used for S-P-F studs but some improvements in kiln efficiency could be attained by configuring kilns as described above. Of more importance would be the extension of high temperature drying for many of the other sizes of this species group.

SECTION

XVIII

INCREASED PRODUCTIVITY WITH OXYGEN ENRICHED COMBUSTION SYSTEMS

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B.C.
13 - 14 April, 1988

Jack R. Ayton
Canadian Liquid Air Ltd.

ABOUT THE SPEAKER:

John R. (Jack) Ayton graduated in mechanical engineering in 1962 from the University of Toronto and joined Canadian Liquid Air in 1963.

He worked on industrial gas applications and technology for 20 years including a period of his career in the production department of Canadian Liquid Air.

Projects included lime and cement kiln enrichment, electric furnace oxy-fuel combustion and reverberatory furnace enrichment in the non-ferrous metals industry as well as cupola air enrichment for cupolas in the cast iron and rockwool industries.

Jack lives in Vancouver and is part of his company's commitment to Western Canada.

Canadian Liquid Air was founded 77 years ago, and has since become the largest producer and marketer of industrial, and specialty gases in Canada.

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DELIVERED BY

JOHN R. AYTON
CANADIAN LIQUID AIR
VANCOUVER, B.C.

The only data neeed to determine burner size and heating time is:

- ladle dimensions - need internal area including cover
- refractory type and/or density
- fuel available
- desired heating time
- wall temperature desired
- ladle use (transport or treatment ladle)

For an economic assesment one needs current

- wall temperature
- fuel consumption
- heating time

CANADIAN LIQUID AIR LTD.

THE ARNOLD MARKEY LECTURE

"INTENSIFIED USE OF OXY-FUEL BURNERS"

During this talk I will endeavour to impart to you a broad understanding of the advantages of oxy-fuel burners which have led to their "Intensified" use.

The use of OFBs began in 1956 in industries where production demands had risen and only small productivity benefits were derived from O_2 enrichment of existing air-fuel burners.

Slide # 2 shows OFBs can replace enrichment in EAF, non-ferrous, glass and refractory melting but both come into their own in other areas. Enrichment in the reheat furnace and cement and lime kilns, burners in induction furnace melting, carbon black production and ladle preheating.

Today, I would like to concentrate on the use of OFBs in EAF melting and ladle preheating.

Slide # 3

I will cover briefly the reasons for transition to EAF melting and furnace improvements before moving on to O_2 use - fusion lancing and oxy-fuel burners, the history, reasons for adoption, advantages they possess and examples of their application.

Then briefly I will show some examples of OFB use for ladle preheating and their advantages.

EAF - Transition to EAF melting

The open-hearth furnace was, along with the Bessemer process, the traditional method of steelmaking. But with the O.H. there were several disadvantages.

. fuel combustion allowed S pick-up from the atmosphere

. the atmosphere was oxidising

. a low temperature was maintained to reduce roof damage from the burner flame.

Despite the cost of electrical power being high in EAF, the production of alloy steels became viable as

. the electrodes produced a reducing atmosphere and

. a much higher temperature was available, hence the alloying elements became less oxidisable than carbon

. flexibility of slag practice (basic or double slag (basic and reducing)) allowed very low sulphurs

. the electrical cost was only a small percentage of the production cost of alloy steels.

Slide # 4

However, with the advent of rapidly increasing fuel oil cost, compared to electricity, and the increasing use of oxygen reducing the refining time, hence, increasing productivity and offsetting the intrinsic costliness of EAF production, steelmakers turned to EAF which now offered modest investment

high flexibility

high productivity and

efficiency

to steel production in general.

Slide # 5

Furnace improvements

As familiarity with the process evolved, and production demands grew, efforts were made to improve existing low power rated steel plant productivity. These improvements

- increased specific power input - transition to U.H.P.
- use of water cooled side wall panels
  roofs
  electrodes
- improved scrap blending
- use of O_2 during melting - fusion lancing

have contributed to the reported 5% per annum increase seen in EAF production.

Slide # 6 shows a typical example of this - a 100 t furnace combining all of the above and the use of oxy-fuel burners. Production has increased from approximately 17t to 69t in a thirty year period.

The use of oxygen is not a new phenomenon but is one which has evolved over the past four decades from

decarburisation lancing (or refining - 1940)

to fusion lancing

and, subsequently, oxy-fuel burners (1956)

The use of oxygen has several advantages, it offers

- production flexibility
- decreases energy costs on a per tonne, or unit production, basis
- low investment
- for capacity gains otherwise unavailable without changes in transformer or melting facilities.

FUSION LANCING

Slide # 7

This is the interreaction of O_2 (supplied via a lance) and combustible (oil on the scrap, carbon in the scrap, an additional carbon charge or any combination of the three).

For 1 lb C + 32 SCF $O_2 \rightarrow CO_2$ + 4.3 kwh (theoretical) practical experience has shown a 65% efficiency of reaction giving 2.8 kwh/lb.C

(1 Kg C + 2 NM^3 $O_2 \rightarrow CO_2$ + 9.5 kwh theory)
(6.2 kwh actual)

The amount of C added is dictated by the

- required final melt chemistry
- amount of oil in the scrap
- %C in the scrap
- required reduction in MDT or substitutional energy

Slide # 8

The benefits to be derived from fusion lancing vary from shop to shop depending upon the practice, steel quality, operators and scrap.

In practice the optimum appears to be 16.6 Nm^3/Tonne (600 SCF/ton) where, in general, the following reductions are available: - 18 mins. MDT, 23 mins T-T, 87 kwh/T (58 kwh/t)

Slide # 9

If one uses less O_2, the benefits are needless to say reduced. This slide shows the results from W.C.S. using 400 SCF/ton total O_2 for reductions of 7 mins in MDT, 32.5 kwh/t.

Another example we have is 351 SCF/t (10.8 m^3/T) with 16 min. reduction in MDT.

Fusion lancing has additional benefits:

. increased bath temperature, for same electrical power

Considering 1 Ton of metal

To remove 1 point of C (0.01%) or 0.2 lbs the heat evolution is 875 BTUs (0.08 kwh)

As 0.21 BTUs will raise 1 lb of molten metal 1°F

i.e. 420 BTU's will raise 1 t of molten metal 1°F

Removal of 0.2 lbs C from 1 ton of steel will raise the bath temperature $\frac{875}{420}$ $\left(\frac{0.08\ \text{kwh}}{0.039\ \text{kwh}}\right)$ or 2.08°F

Thus for 1lb C the temperature rise is 10.4°F

. it can be used in addition to OFBs both during melt down and after a "flat melt" is achieved. In the latter case, the burners are terminated as soon as a "flat melt" is achieved and the lance is installed below the bath level, providing there is C to burn. This further helps as it

. aids diffusion in the melt

. lance pipe consumption may be reduced.

WCS, when they were using 4.64 m^3/T (150 SCF/t), experienced a cost saving of $0.44/Tonne for pipe for the same melt chemistry.

The technique is very simple and not costly as one can use current O_2 piping and supply.

Fusion lancing is often the interim step between decarburisation lancing and OFB.

Fume generation is an additional $7m^3$ fumes/m^3 O_2.

OXY-FUEL BURNERS

The development of oxy-fuel burners began in the mid 1950's. Initial tests performed using air-fuel burners proved unsatisfactory due to the large N_2 "ballast", so oxy-fuel burners were developed.

Slide # 10

O_2 substitution of air, or the removal of the N_2 "ballast" can clearly be seen to

- increase flame temperatures by 820 to 920°C (1500-1700°F) at 100% O_2. The NG curve shows how the temperature rises with increasing % O_2, from 21% (air) to 100%
- permit higher temperatures and hence calorific value for equivalent flame lengths

The flame characteristics, resulting from the internal piping of the burner and the flow characteristics of the fuel and oxygen, give rise to the variation in results obtained from different burner systems.

Adoption of OFBs was much more rapid in Japan than elsewhere. This was due to several factors:

- nearly all their raw material was imported, hence expensive
- industrial sites were rare, existing furnace capacity had to be improved
- most of their furnaces were UHP, negation of cold spots would improve productivity
- electrical energy, made almost entirely from imported crude oil, was expensive.

The use of OFBs allowed the use of cheaper scrap, increased productivity, reduced power costs on a per tonne basis - all of which led to more economical steel production.

It is reported that by 1980 at least 80% of Japanese arc furnaces were equipped, 30% in Europe, whilst in North America it was reported that only 5% had adopted this technology (since then there have been a few door burner trials and installations as well as wall burners - estimated current installations are still very low, less than 10%).

North American non-adoption, in the past, was mainly because of

- lower power costs
- no power retrictions
- steel recession
- fear of - "new technology"
  - alterations in metallurgical characteristics
- retro-fitting an established industry
- furnace modifications - burner ports
  - extraction systems

However, with increased economical productivity being a prerequisite for competitiveness in the current down-turned economic climate, steelworks are considering means to decrease production costs further.

Slide #12 There are 4 major advantages to the use of OFB

1. REDUCED ELECTRICAL CONSUMPTION

Slide # 13 shows the results from a 65 Tonne furnace operating with and without burners. The average KWH/T liquid steel without burners is 540 (470 kwh/t). With O_2 this decreases to 473kwh /T (430 kwh/t) at $16.6m^3O_2/T$ (600 SCF/t).

Slide # 14 shows specific results from a 50T furnace shop running with OFB and fusion lancing. There is a reduction of 70 kwh/T (64 kwh/t) melting and final, which is 16.7% and 11.7% respectively, and an increase of 11.4% productivity for 22m3/T (720 SCF/t) O_2.

Slide # 15 With 4 wall mounted burners, on a 140T water cooled furnace in Italy, the reduction in electrical consumption comparing fusion lancing at $10m^3$ O_2/T (328 SCF/t) with OFBs using $25m^3O_2$/T (820 SCF/t). The melt down kwh is reduced by 100 kwh/T (91 kwh/t) or 23.9%, whilst the reduction for total kwh/T is 90 (82 kwh/t) or 17%.

Slide # 16 a further advantage of OFB is to produce more efficient, faster, melting by

2a. REDUCING THE ARC STABILISATION PERIOD

the lack of arc stability at the start of melting is a mechanical effect due to the variation in charge size (hence height) and coldness. This leads to variable arc lengths being struck as the arc jumps from one piece of scrap to another.

Until the bed is compacted the arc variation continues with the inherent high use of energy and noise level.

Use of OFBs reduces the compaction time, noise level and energy consumption for large sized, less dense charges. With small, more dense charges, such as pellets the burner efficiency is low, so use is not recommended.

More efficiency is obtained when

2b.THE COLD SPOTS ARE REDUCED

Slide # 17

In both low and high powered (>IMVA/Tonne and 70 KA for each electrode) furnaces.

. an isotherm exists indicating hot and cold spots. Around the hot spots there is a higher degree of melting and refractory erosion, whilst the size of the cold spots dictates melting speed. This unequal melt down leads to electrode breakages and unforeseen boiling reactions due to falling scrap.

The oxy-fuel burners reduce the cold spots, allowing more even melting, and increase the overall refractory performance. Combined with water cooled panels the isotherm approaches the furnace circumference.

Slide # 18

In UHP furnaces this thermal pattern is more pronounced and the main reason for OFB use. Initially water cooling was used in the high peak temperature areas but these still remained more pronounced than those in the low powered furnaces. The only solution became OFB use to ensure more efficient melting, though even combined with water cooling the cold spots are never negated.

3. REDUCED FURNACE TIME IS ACHIEVED

Slide # 19

As OFBs reduce MDT and, hence, T-T time by reducing the arc stabilisation period and negating the cold spots. This allows one to

(a) easily increase productivity, on low powered furnaces, by at least 20% (WCS have shown 37.5% to date). In a multi-furnace shop this increased productivity often allows the shop to operate at the same production output with

(b) fewer furnaces or conversely to

(c) reduce the working week.

Slide # 20 shows a comparison between fusion lancing and refining at $15NM^3O_2/T$ (490 SCF/t) versus 3 oxy-fuel wall burners at $25NM^3/T$ (820 SCF/t)

The reduction in MDT is 40 mins (-25%)

and in T-T is 50 mins (-20.8%)

The additional 10 mins saving on T-T compared to MDT is gained in the reduced time required to reach tap temperature and reduced refining time.

Slide # 21

The final advantage is

4. INCREASING AVAILABLE POWER WITHOUT EXPENSIVE TRANSFORMER CHANGES. The input power from OFBs can be 50% of the total power input in low powered furnaces, though, with UHP furnaces it is much less. In the latter case the OFBs are interesting due to the thermal pattern.

BURNER APPLICATION

OFBs are manufactured by a variety of companies who each have their own specifications regarding internal burner design, degree of automation, angle of use, operating power, number of burners to be installed, but the theory remains the same for all.

Firstly the type of OFB system required depends upon the requirements of the prospective operator. The options open are

Door Burners

These are suited for temporary use on furnaces from 5 - 65 Tonnes where production requirements fluctuate and productivity improvements are desired on a temporary basis.

Slides # 22,23,24, 25

The burner is fitted on a trolley, or handled by an hydraulic arm when it is a permanent installation. Door burners are often used to prove the viability of a wall burner system

The range of O_2 used is 15 - 25 m^3O_2/Tonne (490-820 SCF/t) charged, but this is generally $15m^3O_2$/Tonne (490 SCF/t) charged.

Slide # 26

Door burner Operation

Advantages

- reduced MDT 5 - 20 mins
- minimal investment
- no furnace modification
- burner mobility permits angling towards cold spots (change every 10 mins)
- preheats and cuts scrap (reduced fuel ratio)
- in general, requires no modification to-exhaust system
  -refractories
- $1m^3O_2$ 3 kwh

Considerations

- door is occupied, making simultaneous lancing or additions impractical
- generally manual operation
- requires the presence of a full-time operator

Slide # 27 shows the results for a 65T furnace operating a 8.1 MW burner (28M BTU) and a fusion lance - Total O_2 melting 34 Nm^3/T. (1115 SCF/t). Production has increased by 21.4% (i.e. a reduction in T-T of 45 mins) whilst the kwh/T have reduced by 92 (84 kwh/t).

Slide # 28

Wall Burners

Wall burners are suited for permanent use for electrical power substitution and/or increased productivity in furnaces greater than 20 Tonnes. It is possible and desirable to divide the additional heat among several burners, producing greater efficiency when the furnace exceeds a certain size or increased productivity requirement.

Slide # 29

The advantages of wall burner operation are:

- the door is free
- the system can be fully automated
- high thermal input is possible, hence
- MDT is decreased by 15 - 40 mins (40 if fusion lancing is used)
- flexibility increases
- $1Nm^3$ O_2 4.5 kwh

Consideration should be given to

- installing water cooled panels
- whether the exhaust system is capable of handling the increased fumes

Slide # 30 shows an example of benifits to be gained from 3 x 5.8 MW wall burners installed on a furnace with water cooled panels. Column A shows the benefits derived using the same charge weight as without OFB.

The MDT is decreased by 34 MINS, the total kwh/T by 70 (63.6 kwh/t)

for an increase of 27 NM^3 O$_2$/T (885 SCF/t)

-In "B", the charge size has been increased to 78 T (+44.5%) while the T - T and MDT time has increased over "A" by 19 mins, but NOTE the MDT is still 20% less than without the OFBs for 44.5% more production.

The electrode consumption in both cases has reduced by 8.5% and 11.9% respectively.

Having decided which kind of system is to be chosen there are other areas which are considered, to a greater or lesser degree, by all burner manufacturers.

Positioning

Door burners have the flexibility to be angled at the cold spots and moved to cut scrap when and where necessary.

Slide # 31 with wall burners

The wall position is determined from accurate furnace drawings and is dictated by furnace cold spots
door positions
the cold electrode (one is always colder)
tap hole
exhaust system
4th hole position
arc circle
roof support
metal level during tapping
water cooled panels

Generally, they are positioned at 120° seperation at the mid-point between the electrodes(in the cold spots) for maximum efficiency. This also prevents gases impinging upon the electrodes thus decreases electrode consumption.

Slide # 32

Burners are generally in set from the inner refractory wall when in the firing position, to prevent damage from falling scrap.

They are sometimes protected by a special device which protrudes into the furnace. This (slide # 33) burns back" during a campaign but prevents splash back to burner nozzle.

Slide # 34

Water cooling

With the increased thermal input from OFBs it is often necessary or advisable to use water cooling.

Often the burner bodies are water cooled. The water cooling reduces

- the chance of fuel ignition back up the burner
- burner wear
- the slag/steel penetration into the refractories (in the case wall burners)

The water is generally recirculated, filtered and sometimes treated to retard rusting. Water is provided either from an external source or can be supplied in-line with the water cooled panels. However WCS found the latter not to be a good idea and are now converting back to water flow through our panels.

A high thermal energy input is generally available from the burners. If the requirement is such, then it is advantageous to consider water cooling the furnace walls. This can be local slide # 35, 36, 37 a jacket around the burner - for the lower production rates, but, with high production rates, panels or sandwiches, constituting 60-80% and 20-50% of the furnace walls respectively, must be used. (The design and geometry allows for transformer power and production cycle). The use of water cooling allows the use of a long arc, which is more stable, hence more efficient, without increasing the refractory wear (which it would do with warm/hot refractories).

Slide # 38

Refractories

The consumption of the walls is more than that of the roof in normal practice. With burners, there should be no consumption increase until 25 Nm^3 of O_2/Tonne (820 SCF/t) is used. Above this the consumption depends upon refractory quality, specific electrical power and whether or not there are water cooled panels. With water cooled panels a greater energy input can be withstood prior to experiencing wear.

In some technologies the refractories are changed to accomodate the OFBs. Slide # 39 One technology uses a basic plus carbonaceous brick furnace wall which has a

. high thermal conductivity
. resistance to spalling and
. resistance to elevated temperatures

At high temperatures it is not very resistant to slagging or oxidation, hence the furnace walls are water cooled Slides # 40, 41 and jackets are used round the burners.

In general, though, refractory practice remains unchanged and consumption is normally seen to range from the same to 12% decrease on a per tonne basis

Slide # 42

Electrode Consumption

In general this decreases on a per Tonne basis. In the examples seen the decrease is 8.5 to 12%. The reasons for this are burners are not used once a "flat bath" is obtained and, prior to this, the electrodes are protected by the charge. Cold spots are eliminated hence, the amount of falling scrap causing occassional electrode breakages is reduced. In furnaces with water cooled walls the consumption is less than for non-water cooled.

Power Stepping

In our experience, because OFBs reduce arc instability, by rapidly compacting the scrap, it is possible to reduce the number of voltage steps to high power. One customer reduced the number of steps from 6 to 2 with one oxy-fuel door burner. High power is usually used until the desired temperature is obtained then it is reduced to a level which allows one to maintain the bath temperature during refining. However, some manufacturers recommend 2 steps to high power and continuation at high power throughout refining. Adoption of this practice is very much dependent upon the operators, their experience and familiarity with the OFB system.

Slide # 43

Scrap

To further increase OFB efficiency, charge compilation must be repeatable - i.e. the percentage and location of various scrap qualities must be as constant as possible with fine scrap at the base of charges to ensure greater refractory protection. If the scrap is too large, a decrease in heat transfer is obtained, too small and oxidation occurs. Often a large volume of average sized low grade scrap is used. If the %C in the scrap is too high the refining time is increased, if too low refining is impaired.

Slide # 44

Exhaust system

Another major consideration must be the exhaust system. With OFBs, MDT is reduced. This increased melting speed will increase the quantity of fumes/unit time. Further, OFBs produce $9m^3$ fumes/m^3 O_2 additional to those from usual melting. So, for example,

Fumes during normal melting	$1000Nm^3/T$ (36,064 SCF/t)
OFB generally 25 NM^3/Tonne at $9m^3$ fume/m^3 O_2	$225Nm^3/T$ (8,114 SCF/t)
Total	$1225Nm^3/T$ (44,178 SCF/t)

The exhaust system must be capable of providing a relatively constant pressure in the furnace for correct burner flame efficiency. A constant negative pressure allows air to infiltrate, thereby enhancing combustion efficiency. If the negative pressure is too high, reduced thermal efficiency, excessive electrode and refractory wear result. If it is too low, it will lead to excessive fuming.

All the above must be considered in conjunction with logistics and melt chemistry.

Logistics

Increased productivity is only feasible if supplementary steelworks equipment can support increased demands or productivity. (for example charging baskets, ladles, concast, rolling mills), and that the melt chemistry is such that any alloy loss incurred in high alloy steels (or irons) can be withstood in the overall picture.

As burners are directed into the cold zone and use is stopped on obtaining a "flat melt", no melt superheating occurs unless it is specifically desired. Slide # 45 "Problems" are encountered with high alloy steels or oxidisable melts but oxidation is prevented by altering the O_2 - fuel proportions in the burner and controlling the charge size. In this case a fuel rich (less than stoichiometric) mix is used to ensure no O_2 is available for oxidation of alloying elements.

Stoichiometric mixing is used in burners when the scrap is selected free from oil and dust. When oily or high carbon scrap is melted the burners are run O_2 rich to ensure combustion of the oil and/or reduction of the carbon.

If one over compensates on running O_2 rich, the CO produced will lead to secondary combustion with air in the flue. The increased burning of exhaust gas increases the temperature in the flue.

In a steel 13%Mn, 1% Mn is lost by oxidation, in high carbon alloys, the oxidation depends upon the amount of carbon present. In general though, trials have shown a reduction in the cost of alloying element when using OFBs.

Slag Chemistry generally does not change unless the operator requires a change to improve the process. It is very expensive to change slag practice.

Slide # 46

In conclusion, there is a check list to be consulted when considering an OFB system.

One must determine the OBJECTIVES to be gained with such a system, then ensure the LOGISTICS of increased production are there and whether WATER COOLING is necessary or changes are needed to the existing system, as was the case with WCS. The other items,

REFRACTORIES

ELECTRODES

SCRAP TYPE AND MIX

and the EXHAUST SYSTEM are all items to be considered when assessing the ECONOMICS of OFB use with the O2 supplier.

Slide # 47 shows the reductions in MDT and kwh/T that we have found available in practice versus the $Nm^3\ O_2/T$ used. WCS for example previously had a MDT of 130 mins without OFB, with 19.25 $Nm^3 O_2/T$ (630 SCF/t) with OFBs, the new MDT is 105 mins roughly i.e. a 19.2% increase in production.

Slide # 48 Profit is the prerequisite of competitiveness is decreased cost of production.

Slide # 49 shows the production cost breakdown at WCS gained from running 2 burners at 50% capacity (or 7 MW) for this 19.2% increase in production. The costs with OFB total 95.5% of those without OFBs.

As you can see cost of labour is down 18.8%
electrodes down 4.7%
refractories down 8%

the power figure is apparently the same before and after OFBs. However, the cost of power with OFBs is inclusive of the natural gas cost. The actual kwh cost has been reduced by 9%.

Oxygen still remains only 1.3% of the total cost.

The actual cost savings due to OFB are in excess of $4/t. This is not including overhead savings, which are substantial.

LADLE PREHEATING

Another major use of OFBs in the European and Japanese steel industry is for ladle preheating. Because of the hotter flame it is slide # 50 possible to:

- decrease heating time-this is generally halved though the reduction can be greater
- increase wall temperatures to 1300 - 1400°C (2340 - 2550°F)- which is especially advantageous for ladle metallurgy
- increase initial temperatures on ladles to be reheated (by 200-400°C (390 - 780°F)in the case of rapid rotation)

and, as reduced heating time allows accelerated ladle rotation, one can

- reduce the number of ladles - for the same production
- increase production-for the same number of ladles
- save fuel - as there is better combustion efficiency and shorter heating time
- reduce furnace superheating - a practice normally undertaken to compensate for temperature loss due to cold ladles or to allow for ladle treatment or put another way
- save electrical energy in the EAF and
- increase productivity-due to faster furnace turn around
- the other advantage is the lower investment cost as compared to conventional ladle preheaters. To achieve equivalent ladle temperatures it is necessary to install a complex air preheating system for air-fuel burners.

O_2 use is generally 100 - 500 Nm^3/heat (3600 - 18,000 SCF/ht) (from cold) depending upon ladle size and required temperature.

The same arguments apply to ladle heating - the economics are often dictated by the fuel available though hidden savings are reduced heating time (for same wall temperature as obtained with air-fuel burners), and the reduction in ladles.

In order to have greater efficiency the ladles must be covered when using the burners. OFBs cannot be used on stoppered ladles. Burners are installed at 1/2R and protrude into the ladle.

Slide # 51 shows the results of two EAF shops in Europe

in case 1 there is a 75% reduction in heating time which gives 200°C (400°F) higher temperature at 7cm depth in the refractory.

The trade off is

240 kg (1650 SCF) propane and 600 NM^3 (21638 SCF) O_2 for 900 kg HFO (37M BTUS).

Without the OFB the total energy is 10,694 kwh

With OFB it is 6,044 kwh. The burners have saved 7.7 kwh/NM^3O_2 (23 kwh/100 SCF O_2)

in case 2 the heating time is reduced 50% and the temperature is 350°C (650°F) higher at 7cm. The trade off is 280.Nm^3 (10,098 SCF) O_2 for 190 Nm^3 (662 SCF) N.G.

Total energy input without OFB is 3,487 kwh, with OFB it is 1,395 kwh.

The savings are 7.4kwh/Nm^3 O_2 (22kwh/100 SCF O_2).

#1

INTENSIFIED USE OF OXY-FUEL BURNERS

#2

Application	O_2 Enrichment	Oxy-Fuel Burners
EAF Melting	✓	✓
Ladle Preheating	—	✓
Reheat Furnace	✓	—
Non-Ferrous Melting	✓	✓
Glass Industry	✓	✓
Refractories	✓	✓
Cement & Lime Kilns	✓	—
Induction furnaces	—	✓
Carbon Black Production	—	✓

#3

INTENSIFIED USE OF OXY-FUEL BURNERS

- EAF
    - TRANSITION TO EAF MELTING
    - FURNACE IMPROVEMENTS
    - FUSION LANCING
    - OXY-FUEL BURNERS
        - HISTORY
        - ADOPTION
        - ADVANTAGES
        - APPLICATION
- LADLE PREHEATING

#4

TRANSITION TO EAF MELTING

- MODEST INVESTMENT
- HIGH FLEXIBILITY
- HIGH PRODUCTIVITY
- EFFICIENCY

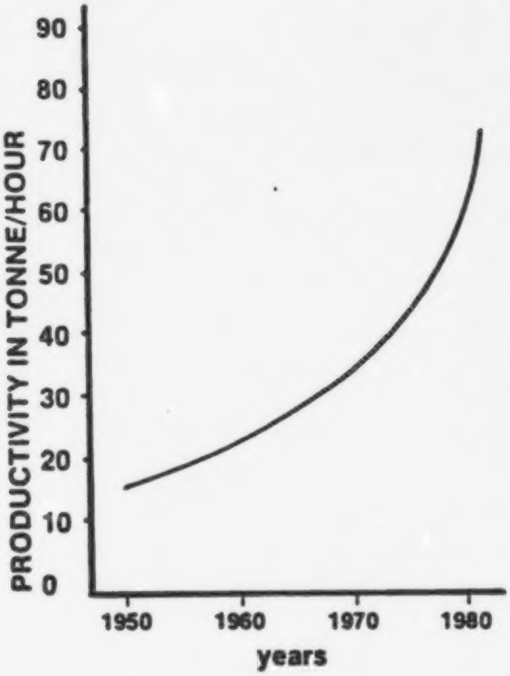
#5

E.A.F. PRODUCTIVITY

MAIN IMPROVEMENTS OVER PAST 2 DECADES

- INCREASED SPECIFIC POWER INPUT
- USE OF WATER COOLED SIDE WALL PANELS
  ROOFS
  ELECTRODES
- IMPROVED SCRAP BLENDING
- USE OF O_2 DURING MELTING

#6



DEVELOPMENT OF THE PRODUCTIVITY OF A TYPICAL ARC FURNACE

#)

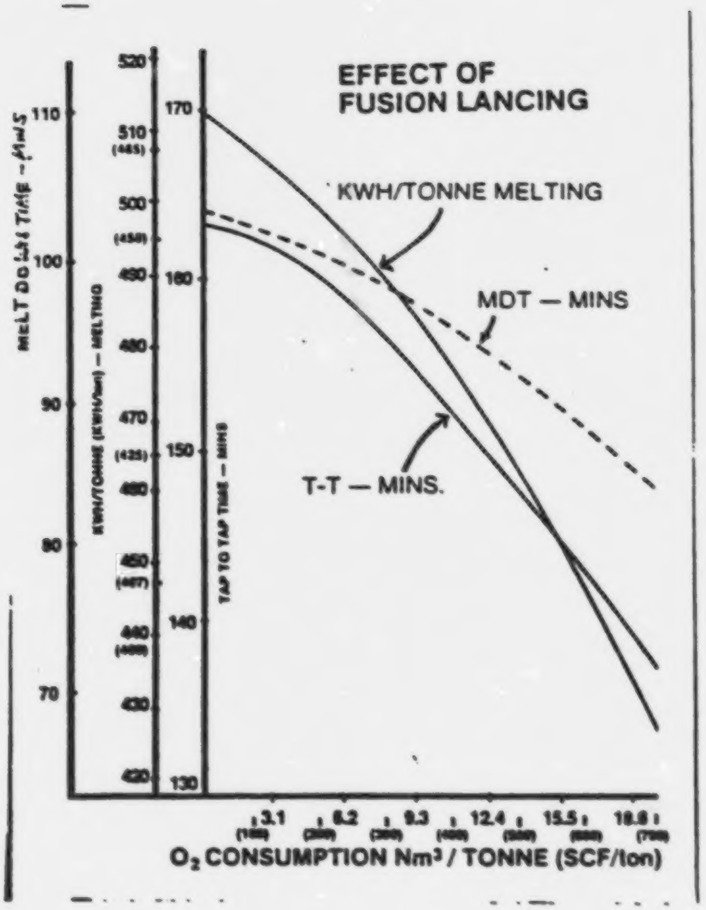
FUSION LANCING

$$C + O_2 \longrightarrow CO_2 + \text{HEAT}$$

CARBON ADDITION IS DICTATED BY:

- REQUIRED FINAL MELT CHEMISTRY
- AMOUNT OF OIL IN SCRAP
- % C IN THE SCRAP
- REQUIRED REDUCTION IN MDT OR SUBSTITUTIONAL ENERGY

#8



EFFECT OF FUSION LANCING

#9

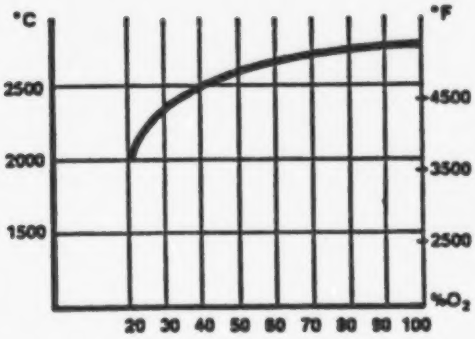
WESTERN CANADA STEEL
FUSION LANCING

	WITHOUT	WITH	
TAP TO TAP TIME - MINS	137	130	(-5%)
MELT DOWN TIME - MINS	90	83	(-7,8%)
REFINING TIME - MINS	30	15~~20~~	(-50%)
POWER KWH/T - MELTING	530	497.5	(-6%)
- TOTAL	420	387.5	(-7.7%)
O_2 SCF/T - REFINING	50		
- FUSION	0	400	
ELECTRODE CONSUMPTION - LBS/t	13.9	12.6	(-9.4%)
DAILY PRODUCTION AVE.	280	294	(+5%)

#10

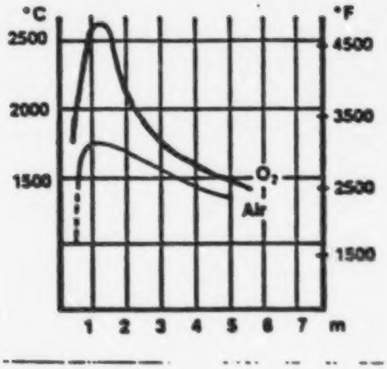
- Theoretical flame temperature achieved by oxygen enrichment.

	Air	Oxygen
Natural gas	1950 °C	2780 °C
Propane	1990 °C	2830 °C
Butane	1916 °C	2780 °C
Mapp gas	2090 °C	2915 °C
Coke-oven gas	2010 °C	2830 °C
Coal	2010 °C	2930 °C



#11

- Temperature profile along flame axis (air and oxygen respectively).

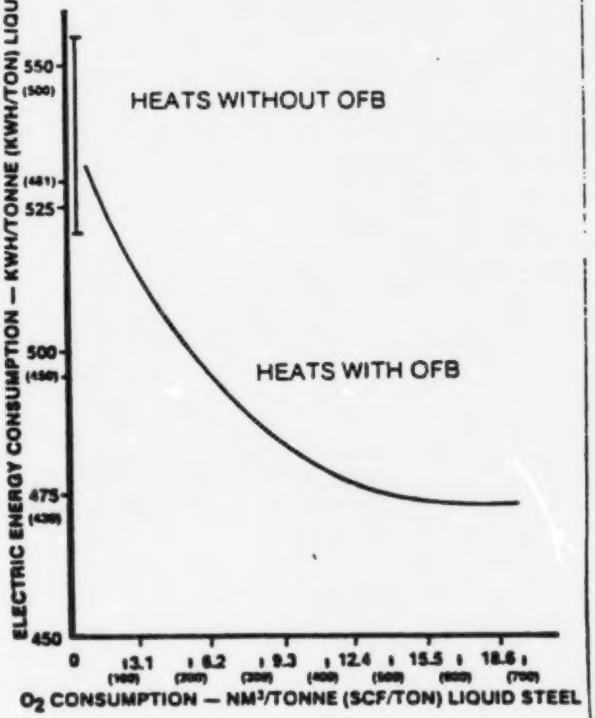


#12

ADVANTAGES OF OXY-FUEL BURNERS

1. REDUCE ELECTRICAL CONSUMPTION

#13



INFLUENCE OF OXY-FUEL BURNER OPERATION ON ELECTRICAL ENERGY CONSUMPTION

#14

1 BURNER THROUGH DOOR (GERMANY)

FURNACE CAPACITY - 50 TONNES
CHARGE WEIGHT - 52 TONNES
TRANSFORMER CAPACITY - 15 MVA

	WITHOUT	WITH BURNER
TAP TO TAP - MIN.	175	150-160 (-11.4%)
MELT TIME - MIN.	110	90 (-18%)
BURNER TIME - MIN.	—	30
CONSUMPTIONS		
O_2 Nm^3/T - MELTING	—	15 BURNER 7 LANCE
- REFINING	5	5
NATURAL GAS - Nm^3/T	—	7 + 250 KG COAL
ELECTRIC KWH/T - MELTING	**420**	**350 (-16.7%)**
- TOTAL	**600**	**530 (-11.7%)**

#15

4 WALL MOUNTED BURNERS — WATER-COOLED PANELS (ITALY)

FURNACE CAPACITY - 140 TONNES
CHARGE WEIGHT - 150 TONNES
TRANSFORMER CAPACITY - 75 MVA

	WITHOUT	WITH BURNER
TAP TO TAP - MIN.	177	157 (-11.3%)
MELTING TIME - MIN.	92	72 (-22%)
REFINING TIME - MIN.	60	60
REPAIR AND CHARGE - MIN.	25	25
PRODUCTIVITY T/HOUR	41.5	46.6 (+12%)
CONSUMPTIONS		
ELECTRIC KWH/T - MELTING	**418**	**318 (-23.9%)**
- REFINING	**83**	**83**
- TOTAL	**538**	**448 (-17%)**
O_2 Nm³/T - MELTING	10	25
- REFINING	5	5
NATURAL GAS - Nm³/T	—	10

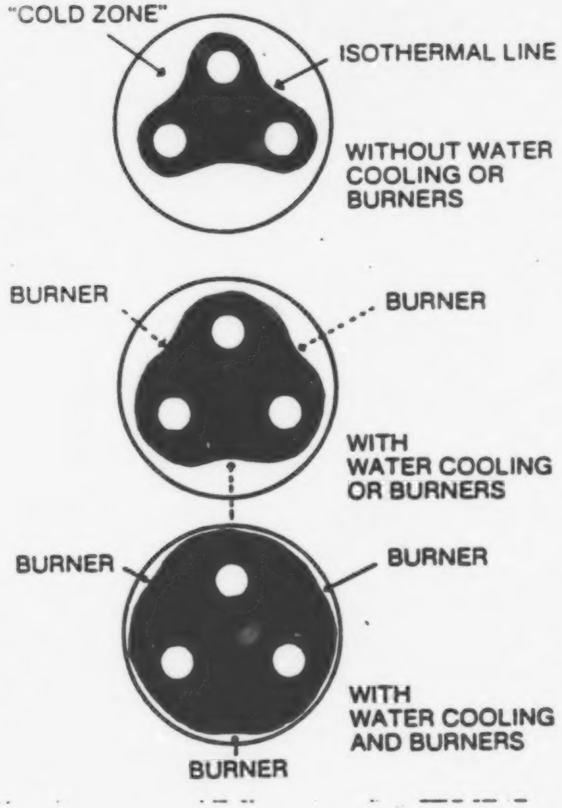
#16

ADVANTAGES OF OXY-FUEL BURNERS

1. REDUCE ELECTRICAL CONSUMPTION
2. PRODUCE MORE EFFICIENT (FASTER) MELTING BY:
   - A) REDUCING THE ARC STABILISATION PERIOD
   - B) REDUCING THE COLD SPOTS

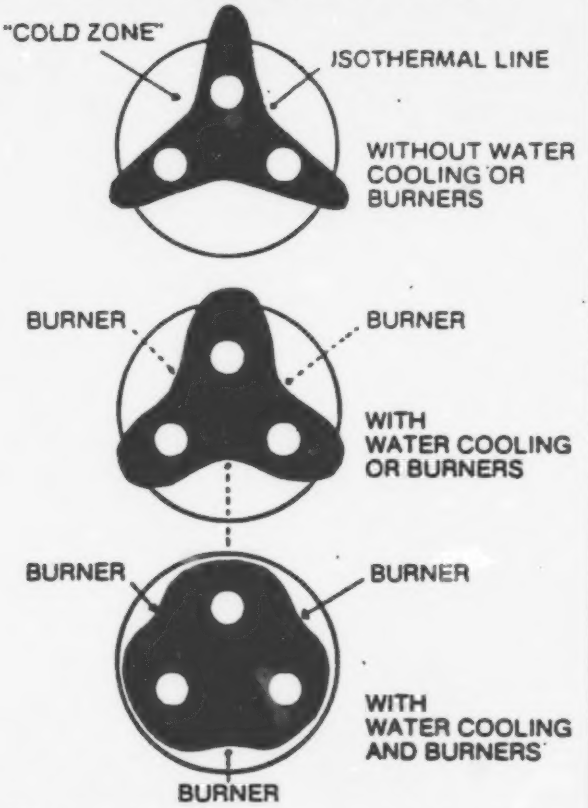
#17

DECREASING COLD ZONES IN
LOW POWER FURNACES


"COLD ZONE"
ISOTHERMAL LINE
WITHOUT WATER COOLING OR BURNERS
BURNER
BURNER
WITH WATER COOLING OR BURNERS
BURNER
BURNER
WITH WATER COOLING AND BURNERS
BURNER

#18

DECREASING COLD ZONES IN ULTRA HIGH POWER FURNACES


"COLD ZONE"
ISOTHERMAL LINE
WITHOUT WATER COOLING OR BURNERS
BURNER
BURNER
WITH WATER COOLING OR BURNERS
BURNER
BURNER
WITH WATER COOLING AND BURNERS
BURNER

#19

ADVANTAGES OF OXY-FUEL BURNERS

1. REDUCE ELECTRICAL CONSUMPTION
2. PRODUCE MORE EFFICIENT (FASTER) MELTING BY:
   - A) REDUCING THE ARC STABILISATION PERIOD
   - B) REDUCING THE COLD SPOTS
3. **REDUCED FURNACE TIME PERMITS:**
   - **A) INCREASED PRODUCTION**
   - **B) DECREASED NUMBER OF FURNACES OPERATING**
   - **C) REDUCED WORKING WEEK.**

#20

3 WALL MOUNTED BURNERS (ITALY)

FURNACE CAPACITY - 70 TONNES
CHARGE WEIGHT - 90 - 95 TONNES
TRANSFORMER CAPACITY - 24 MVA

	WITHOUT	WITH BURNER
TAP TO TAP-MIN.	**240**	**190 (-20.8%)**
MELT TIME-MIN.	**160**	**120 (-25%)**
REFINING TIME - MIN.	45 - 50	45 - 50
CONSUMPTIONS		
ELECTRIC KWH/T		
-TOTAL		535
-MELTING	570-590	
O_2 - Nm^3/T	15+ 1% Anthracite	25
PRODUCTION - T/D	510	595 (+17%)

#21

ADVANTAGES OF OXY-FUEL BURNERS

1. REDUCE ELECTRICAL CONSUMPTION
2. PRODUCE MORE EFFICIENT FASTER MELTING BY:
    - A) REDUCING THE ARC STABILISATION PERIOD
    - B) REDUCING THE COLD SPOTS
3. REDUCED FURNACE TIME PERMITS:
    - A) INCREASED PRODUCTION
    - B) DECREASED NUMBER OF FURNACES OPERATING
    - C) REDUCED WORKING WEEK
4. **INCREASE AVAILABLE POWER WITHOUT TRANSFORMER CHANGE**

#26

DOOR BURNER OPERATION

ADVANTAGES

- REDUCED MDT 5-25 MINS
- MINIMAL INVESTMENT
- NO FURNACE MODIFICATION
- BURNER MOBILITY
- PREHEATS AND CUTS SCRAP
- NO MODIFICATION TO EXHAUST SYSTEM
- $Im^3\ O_2 \longrightarrow 3$ KWH

CONSIDERATIONS

- DOOR OCCUPIED
- GENERALLY MANUAL OPERATION
- REQUIRES FULL-TIME OPERATOR

#27

1 BURNER THROUGH DOOR (SPAIN)

FURNACE CAPACITY - 65 TONNES
TRANSFORMER CAPACITY - 8 MVA

	WITHOUT	WITH BURNER
TAP TO TAP - MIN.	210	165 (-21.4%)
MELT TIME - MIN.	115	75 (-35%)
BURNER TIME - MIN.	—	60
REDUCTION IN KWH/T	—	92
CONSUMPTIONS		
O_2 Nm^3/T - MELTING	—	14 LANCE 20 BURNER
- REFINING	5	5
FUEL L/T	—	9.2 HEAVY FUEL OIL 10.8 KG COAL

#.29

WALL BURNER OPERATION

ADVANTAGES

- DOOR IS FREE
- CAN BE AUTOMATED
- HIGH THERMAL INPUT IS POSSIBLE, HENCE
- MDT DECREASES 15-40 MINS
- FLEXIBILITY INCREASES
- INm^3 O_2 ⟶ 4.5 KWH

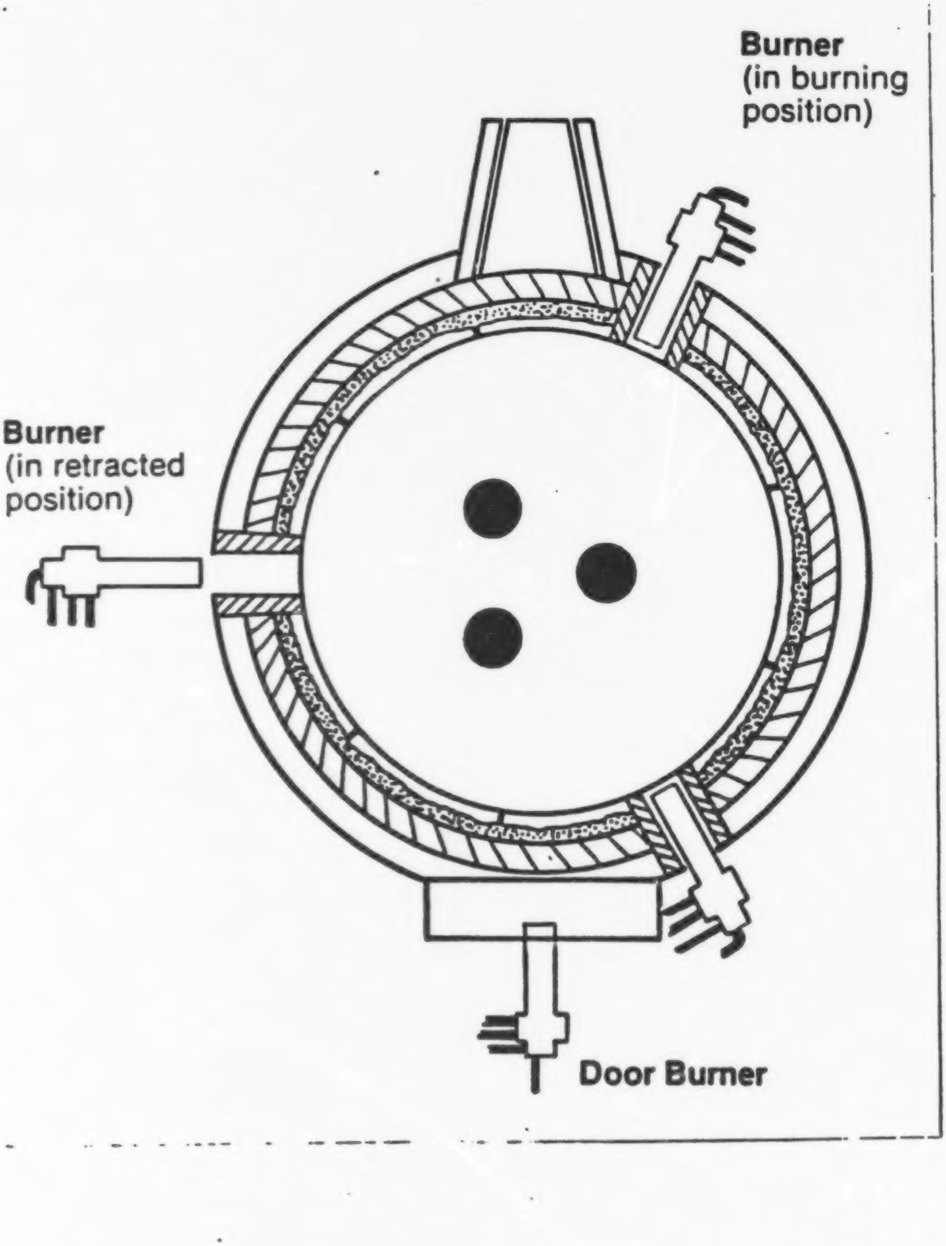
CONSIDERATIONS

- WATER COOLED PANELS
- EXHAUST SYSTEM

3 WALL BURNERS — WATER COOLED FURNACE WALLS (SPAIN)

	WITHOUT	WITH OFB	
		(A)	(B)
NOM. FURNACE CAPACITY - TONNES		45	
CHARGE WEIGHT - TONNES	57	57	78 (+44.5%)
TRANSFORMER CAPACITY - MVA		24 (+20%)	
TAP TO TAP - MIN	?	110	129
MELT TIME - MIN	90	56 (-37.8%)	75 (-20%)
ELECTRIC KWH/T - MELTING	?	370	
- TOTAL	500	430	
TOTAL O_2 - Nm^3/T	16	43	
ELECTRODE CONSUMPTION - KG/T	5.9	5.4 (-8.5%)	5.2 (-11.9%)
WALL	MGO	GRAPHITE + MGO	
BURNER		5.8 MW	x 3
BURNER TIME - MIN		24	

#31


Burner
(in burning
position)
Burner
(in retracted
position)
Door Burner

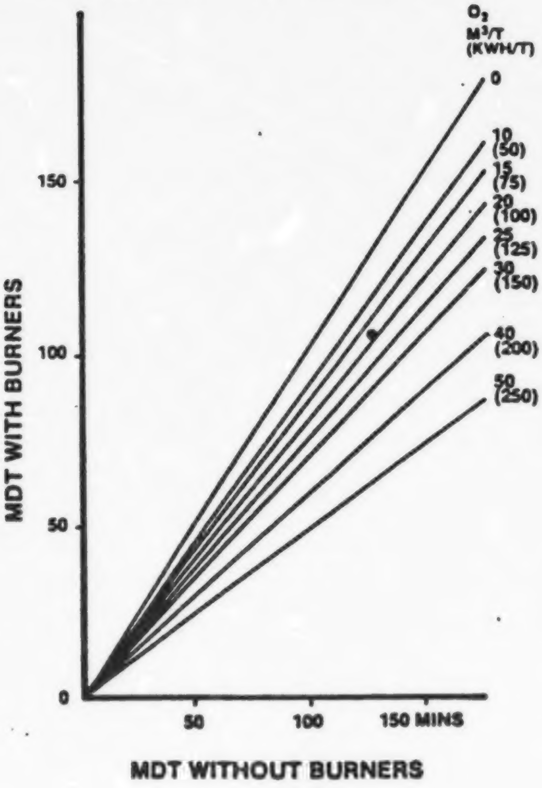
#46

OFB CHECK LIST

- OBJECTIVE
- TYPE OF SYSTEM
- LOGISTICS
- WATER COOLING
- REFRACTORIES
- SCRAP
- MELT CHEMISTRY
- EXHAUST
- ECONOMICS

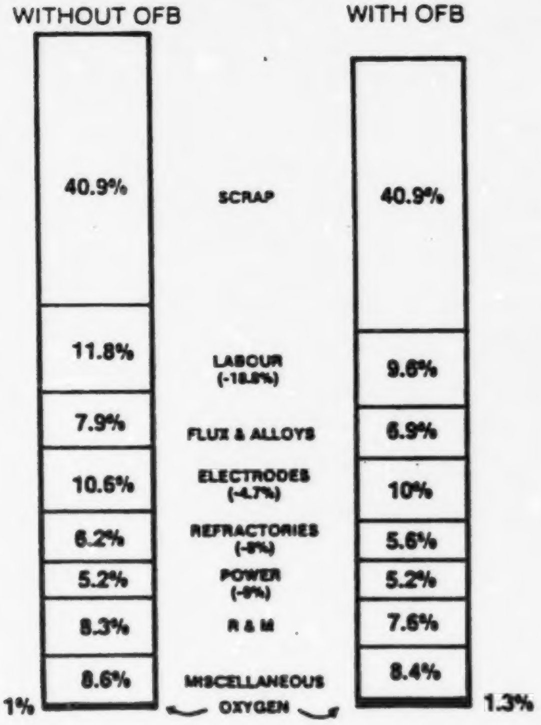
#47

OXYGEN VERSUS DECREASE IN MDT



#49

PRODUCTION COST BREAKDOWN
W.C.S.



#56

OFB ADVANTAGES
- LADLE PREHEATING

- decreased heating time
- increased wall temperatures
- increased initial temperatures
- reduced ladle stock, or
- increased production
- reduced fuel consumption
- reduced superheating, hence
- reduced electrical consumption
- increased productivity
- low investment cost

#51

LADLE PREHEATING — OFB

	CASE 1		CASE 2	
	WITHOUT	WITH	WITHOUT	WITH
LADLE SIZE — TONNE	80		20	
HEATING TIME — hr	16 - 17	4	5	2.5
TEMP. °C — 7 cm	500	700	800	1,150
— surface		1,328		1,200
CONSUMPTIONS — fuel	900 HFO	240 kg propane	350 Nm³ NG	160 Nm³ NG
— O_2	—	600 Nm³	—	280 Nm³
TRADE	900 HFO - 240 kg propane - 600 Nm³ O_2		190 Nm³ NG for 280 Nm³ O	

SECTION

XIX

INFRARED IN THE PULP AND PAPER INDUSTRY

Presented at
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ABOUT THE SPEAKER:

Mr. Ross Galbraith has been involved in the Pulp and Paper Industry for the past 25 years.

First in Canada with Broughton Corporation which is now Albany Engineered Services in Glenn Falls, N.Y. He then got involved in infrared drying methods and joined Techmark organization in Newport, R.I. (1981-1985). 1985 to present he has been with Krieger Corporation as Marketing Manager and then General Manager. Krieger Corporation is the U.S. operations of Krieger GmbH in West Germany.

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INTRODUCTION

Electromagnetic energy, often called radiant energy, is propagated by wave motion and exists in all wavelengths between 0 and infinity. Most of the usable radiant energy occurs in the extremely varied properties and produce a great variety of effects. We will look at them briefly. The energy band between 0.4 and 0.75 microns is visible as a red light and wavelengths above 0.75 up to approximately 400 microns become invisible and are known as infrared energy.

There are many types of infrared generators in this range. However, the major portion of the energy which we will discuss today, is emitted in the 1 to 6 micron band, which happens to be the major absorption band of most materials, including water.

INFRARED GENERATORS

The interest in infrared development over the past 30 or 40 years has resulted in many types of infrared generators. Basically, they can be divided into two general classifications, by the fuel used: Gas and electricity. The gas-fired generators fall into five general categories:

1. Radiant Type Burner: where the air/gas mixture is fired into the tube, the surface of which becomes the radiator.

2. Pre-mix Burner: such as the spiral flame, which as the result of controlled velocity, spreads the flame over a large, flat surface, generating radiant heat or infrared.

3. Surface Combustion Burners: which utilize a porous refractory or matrix for the radiating surface on which the gas-air mixture burns creating the radiator. Into this same group would fall drill port ceramics and metal screens.

4. Direct-fired Refractory Burners: utilize the impingent of the gas flame on the refractory and produces the highest intensity, flux density and radiant.

5. Catalytic Burners: that use a catalyst to oxidize gas at low temperatures.

The equipment just described would fall into the temperature range of approximately 1400 to 2100 degrees F or 800 to 1150 degrees C.

The electrical emitter falls into a single category where a Tungsten element burns in a halogen filled tube at up to 4000 degrees F or 2200 degrees C.

DESIGN CRITERIA

Let's look at some of the problems a designer faces in developing an infrared drying system specifically suited for the drying of paper and board products, as well as the drying of coatings applied to these products at various stages of production.

The heart of any IR drying system is the infrared radiation emitting unit, which we will simplify and call an emitter. It has to offer the right compromise between power output and energy absorption by water. Some of the key properties of IR energy important to the paper maker move in opposite directions as the temperature of the emitter is varied: the power output of the emitter increases with a rise in temperature, while the absorptivity of that energy by water generally decreases with a rise in temperature. The higher the temperature, the shorter the wavelength.

The emitter has to be specifically designed for the hostile ambient conditions in a paper mill. Emitter reliability is an important design criterion, for any failure results in unwanted and expensive down time, or at best, a loss in production.

The emitter, as well as the entire system, has to be designed for maximum energy efficiency. Since operating costs of any drying system are vitally important, the system has to operate with the lowest cost possible, and must have the highest possible efficiency in the conversion of the fuel source in calorific value in the radiant energy.

The exposure of paper, board, and especially of coatings to the high energy densities of the IR system has to be designed with a number of unique features that will ensure that these parameters will be enhanced. The purpose for the installation of any IR system is to either increase production, or improve quality, or to accomplish both.

Above all, the system has to be designed for maximum safety. Regardless of the type of emitter used, be it electric or gas, each presents a potential hazard that must be recognized, and system design must be such that these hazards be eliminated.

THEORY

The first slide shows how infrared energy is calculated for any given emitter.

Slide 1 - Power Output Formula $Q = (T_1 4 - T_2 4)$

Slide 2 - Q = Power Output of radiation

Q = Total amount of radiation leaving a radiating surface per unit of time

Slide 3 - A = Incandescent surface area

This is a geometric area of the radiation emitting substance that is actually radiating in the direction of the sheet. (Not to be confused with overall size.)

Slide 4 - Stephan-Boltzmann Constant

A mathematical constant required to determine a value/or Q in Watts (5.67 x 10-8).

Slide 5 - E = Emissivity of radiating substance.

This is defined as the ratio of the emitting substance of that of a black body. The ideal radiator with an E of 1.0.

Slide 6 - $T_1 4$ = Temperature of radiation substance in degrees Kelvin.

$T_1 4$ = Temperature of radiation receiving substance (the sheet) in degrees Kelvin.

Slide 7 - Energy transfer into sheet or coupling energy.

Q - (a + b + c + d) = Coupling energy

a = Radiated energy intercepted by substances before it reaches the sheet

b = Radiated energy reflected by the sheet

c = Radiated energy that is transmitted through the sheet

d = Radiated energy emitted but not toward the sheet

Some manufacturers tend to underplay the value of emissivity. Let's examine just how important a part it plays. Assume the following:

Emitter "A" has an E of .88 and Emitter "B" an E of .72. These numbers are characteristic of a metallic and a ceramic emitter in common use today.

Emitter "A" is operated at 1050 degrees C and generates 3.3 watts/hr from its 150 x 200 mm surface. The wavelength at this temperature is 2.19 microns.

Emitter "B" (same surface area), because of its lower emissivity, must operate approximately 100 degrees C higher in temperature to produce the equivalent output in watts/hr. Because of the higher temperature the wavelength would decrease to around 2.03 microns. This immediately puts Emitter "B" at a disadvantage for two reasons:

1. Its higher operating temperature will substantially reduce its useful life.

2. The shorter wavelength moves it further away from the desired point where infrared is most effectively coupled with water.

Slide 8

This slide shows that these are major absorption points at around 3 and 6 microns. A minor point occurs at just under 2 microns.

These three colours beginning at the top with green are for a 5 micron layer of water. The two others, red and blue, are for 10 micron and

20 micron layers respectively. These three water layers are representative of moistures found in some coating applications.

Slide 9

If 3 microns are a good place to be then why not design an emitter for that wavelength? First, we convert 3 microns into degrees K by dividing the wavelength 3 into 2898 giving us 966 degrees K = 698 degrees C = 1279 degrees F.

Power output at this temperature:

.72	1850 BTU/hr (0.055 kW)
.88	2605 BTU/hr (0.76 kW)

After looking at these two power outputs at the relatively low temperature of 1279 degrees F, it becomes apparent why infrared emitter design is a compromise.

HOOD DESIGN

Slide 10

Infrared drying systems can be enhanced, with the use of circulating air. Much of the waste energy in the hot exhaust gases can be recirculated to improve efficiency. A properly designed system should not only have well designed circulation air, but should also include automatic sheet temperature control, especially in cases when the drying of coatings is involved. Too high a surface temperature can cause overheating of the coating surface resulting in skimming or blistering. Automatic sheet control can maintain sheet temperature as low as 200 degrees F (93 degrees C). Another important reason for circulating air is to remove the vapour between the sheet and the emitter. A third important reason is that a well designed system will have air nozzles that create the micro turbulences that break up the boundary layer to permit more efficient heat transfer.

Circulation air, to be effective must be sized to handle a minimum of 200 cfm per square foot of emitter surface at ambient temperature. The air can be blown over a single row of emitters before it is sucked up into the exhaust duct or over a double row. Hoods can be built as a single row, double row or any combination of both dependent on design criterion or as space limitations dictate.

Retraction systems are custom built to suit the machine location as well as the application. This can be a simple clam shell configuration to permit threading on the tending side or complete retraction when needed. The most common means of motivation are jack screws and pneumatic cylinders.

INFRARED DRYING APPLICATIONS

Beginning from the wet end of the paper machine, the most common locations are as follows:

1. hot pressing;
2. preheating;
3. incremental drying;
4. coating drying;
5. size press;
6. saturator;
7. profiling.

HOT PRESSING

Although not commonly used in this country, hot pressing can be very cost effective. Raising the water temperature to lower viscosity and assist in dewatering the sheet can be done either before or between the press sections. Messrs. Radwan and Nayar of Domtar Research have done impressive testing in this area. The dispute as to whether to steam boxes or infrared systems are most effective should be left until the obvious question is answered: does the mill have either a supply of waste steam, or can it be produced at a lower cost than natural gas? No doubt there is some validity in the claim that the use of IR in the press section, as contrasted to steam boxes, is the penetration without the limitations of conduction of heat released in the condensation of steam.

PREHEATING

Preheating between the last press and the first can dryer is one application that is ideally suited to an IR system. When the sheet temperature is raised to about 180 degrees F or 82 degrees C, a chain reaction generally occurs. With the evaporation process already begun, the sheet wraps the first dryer can with a microscopic vapour film on it surfaces thus eliminating picking. This permits number one dryer to be operated with a surface temperature of up to 215 degrees F or 100 degrees C. Dependent on the grades, this could result in a production increase of 10% to 20%. The main dryer section, relieved of its reheat load, can now take full advantage of the pocket ventilation system. This, coupled with the fact that there is less surface temperature differential front to back on the first few cylinders, may have a positive effect on caliper variations. In turn this has resulted in a much improved moisture profile at the reel. Some, or all of these results have been documented in preheat installations.

Slide 11

Wet end installation facts.

1. As sheet temperature increases water viscosity decreases. More even dewatering.

2. As steam cylinder temperatures in preheat section are increased, their temperature variation front to back is decreased.

3. A preheated sheet (180 degrees F) will carry with it a microscopic vapour film that acts as a lubricant. This helps to eliminate picking.

4. With preheat section actually evaporating at the constant rate, pocket moisture is spread out over the main dryer section.

These four events combine to reduce CD moisture variations out of the main dryer.

INCREMENTAL DRYING

Incremental drying, as the name implies, is IR drying any place on the paper machine when it is feasible to introduce additional energy. Breaker stacks, open draws, even on top of dryer cylinders can be acceptable. The two prerequisites are: the sheet must be at evaporation temperature and must also be in the constant rate zone, i.e. more than 20% moisture level.

COATING DRYING

Slide 12

By far, the largest majority of infrared drying systems throughout the paper industry are used in coating drying. Ideally, the drying of coatings using infrared requires the use of an air circulation system. The air is blown in the sheet direction over two rows of emitters and is taken up through the suction side of the air system to the fan. At this point, a portion, usually 25% or more, is dumped to atmosphere and the cooler make-up air is mixed with the remaining hot air and brought back to scrub the sheet once again. This actually has a number of beneficial effects.

1. By varying the amount of make-up air you can control the sheet temperature to below 200 degrees F.

2. The moisture driven off by the emitters is carried away from the machine.

3. In the event of a break, the air can be used to keep loose break from blowing up into the face of the emitters, at the same time assisting in the cool down of the emitters.

HOOD CONFIGURATION

Hood configuration can vary greatly in coating drying from single to multiple rows of IR. On a new paper machine, or a rebuild of an existing machine, the selection of rows and their spacing can be custom made to suit the product mix as well as the coating applications and speeds. In most cases, however, IR systems in the mills in North America, are fitted into existing space parameters.

Slide 13

Example: Boxboard on-machine coater with two coating stations.

Basis Weight: 130#/3000

Sheet Moisture: 3% into Station 2

Present Speed:	1100 fpm out of Station 1	Sheet Temperature:	130 degrees F
Production:	200,000 tpy	Coating Applied:	7#/3000 - 58% solids
		Sheet Moisture:	3% into Station 2
		Sheet Temperature:	140 degrees F out of Station 2
		Coating Applied:	6#/3000 - 58% solids
		Sheet Moisture:	5% at the reel

Objective: Increased production by 20%.

Set the coating with infra-air at 75% solids in each station.

Recommend: Install 6 feet infra-air immediately following first station.

Install 5 feet infra-air immediately following second station.

Investment: $550,000

Results: 20% production increase or 40,000 tpy.

Energy costs: Depend upon efficiency of drying system following. Total energy efficiency would be raised to between 50% and 80% subject to specific design integration.

Payback: Almost certain less than 2 months.

SIZE PRESS

In the highly competitive paper market today, paper makers are constantly struggling to obtain every ton of production from their machines. Frequently, the bottle neck is located at the size press. A small infrared drying system can give a sizeable increase in production, although the solids content at a typical size press application is generally quite low, so is the percentage of solids, and the amount of water that has to be removed can be considerable.

Slide 14

Example: Paper

Basis Weight:	30#/300	Sheet Moisture:	3% into Station 2
Present Speed:	2800 fpm	Sheet Moisture:	33% out of Size Press
Production:	107,000 tpy	Sheet Moisture:	5.5% at the Reel

Recommend:	Install 6 feet of infra-air immediately following the size press on an open draw and first top cylinder.
Investment:	$400,000
Results:	15% increase.

Temperature of sheet increased past 170 degrees F.

Production increase of 16,000 tpy.

Energy costs: $215,000 per year, or less than $14 per ton added production.

Payback: Likely to be less than 2 months.

SATURATOR APPLICATIONS

Saturators normally present an ideal application for infrared drying. Usually the sheet is coming out of the saturator into a vertical run. This gives the mill the opportunity of locating infrared on both sides of the sheet. Normally, if a longer vertical draw is required, it simply means the elevation of the first paper roll. In many cases saturated drying is a fairly simple operation due to the fact that the sheet is running at slow speed, therefore, the dwell time over the face of the emitters is much longer than usual and, consequently, to set the coating prior to going over to the first paper roll, the IR system must not be long in the MD direction.

PROFILERS

The hood configuration on profilers is normally done in increments of two rows. One of the prerequisites to any profiling system is its ability to turn down from 100% to zero. In the initial stages of profile development, the only suppliers who could accomplish this were the manufacturers of electric infrared. Such is not the case today. The input of even a small amount of energy (say 25%) where ideally it should be zero, has a very detrimental effect on the accuracy of the total system.

Most paper machines have about five times greater CD moisture variations than MD disturbances. The potential savings associated with cross directional control can be very large. It is possible, and is actually being done today, to control to 1/2% at the reel, with present state-of-the-art equipment. Typical savings are in the range of $3/4 million for speed-limited machines and probably in excess of $1-1/2 million for dryer limited applications.

For many grades, the quality improvements are invaluable. For example, with newsprint, it has been estimated that up to 75% of all unexplained press breaks are caused by poor moisture profiles.

On-Line moisture scanners can create some problems of measurement of CD moisture variations which will have to be considered in applying controls.

Measurements are made at a frequency of about 50 per second. These measurements are then averaged for each slice group position or about 6 inches of the CD scan. During each 6 inches travel in the cross direction, the sensor is subject to about 100 times more MD disturbances. In order to get reliable updates of the CD profile, many scans have to be made and averaged over time. The average profile displayed on the CRT is therefore subject to considerable time delay and two types of arithmetic averaging.

In view of these considerations, CD control will have slow response and manual operations would be important in the start-up mode. It is also an advantage if the actuator does not respond too quickly to change. It is the opinion of most major manufacturers of on-line computer equipment today that the speed of the actuator is unimportant. It is the ability of the on-line computer to give the precise signal to the actuator which counts.

The limitations of measurement and actuator performance must be clearly understood. Each actuator has a certain amount of coupling with adjacent units. This coupling phenomenon makes it very difficult to achieve precise controls at the slice and with variable crown rolls in which an adjustment in one direction creates an opposing mode on each side of the actuator.

No coupling between actuator creates an even greater problem. The ideal degree of coupling is an overlap of approximately 20% and must be known in the selection of the appropriate control algorithm.

Some machines will have several different CD controls: the slice screw adjustment controls basis weight which automatically affects the moisture profile, steam showers, water sprays, variable crown rolls, induction heated rolls or dryers are some of the CD controls which interact with a profiling dryer. The degree of interaction must be clearly defined and incorporated into the control system.

Slide 15

Dry end installation facts.

1. Regardless of drying method used (steam can, air of IR) the BTU requirement to evaporate one pound of water increases significantly the dryer the sheet becomes. With IR drying this deterioration occurs later, but it does occur.

2. In the process of removing bound water, significant internal vapour pressure must be created.

   Result: Increasingly the IR energy is absorbed by the fibres.

3. As the fibres absorb more energy the efficiency of the IR system decreases.

   Result: More energy must be applied.

4. When IR is applied in the profile mode just prior to the reel, significant temperature differences will develop in the CD.

   Result: Calendar problems.

   Reel building problems.

   Uneven temperature profile can carry over into on-machine coater.

TRADITIONAL METHODS

There are many CD control approaches which have been utilized with varying degrees of success. Some other traditional approaches to CD controls are slice screws, steam boxes, variable crown rolls.

It is our opinion that all of these methods should be used, or at least be considered, before a dedicated profiler is installed on a paper machine.

INFRA-FLOAT

A revolutionary type of drying system that has made its way onto the market recently is a combination of an infrared coating dryer system and a revamped floatation dryer. Let us assume that an IR dryer has been sized to dry the coating colour to the immobilization of gel point. The exhaust gases in this case around 285 degrees F. This latent heat, which would normally be exhausted to atmosphere, is fed into the Infra-Float through its own fan and heat exchanger system. The result is a combination which can have a combined efficiency of up to 75%. A small auxiliary gas burner is part of the Infra-Float apparatus which supplies additional energy if and when it is called for. A pilot unit has been in operation in a European mill for almost 2 years and a second system will be started up in the U. S. by September of this year.

Some pulp machines have reached their production capacity. Further speed-up can only be attained by major capital expenditures if they go the conventional route. The alternative is the use of IR. Probably one of the most difficult decisions a paper maker has to make is to deviate from the normal production processes and go with a new technology. To any of you who find yourselves in that position, you will find both the gas industry, as well as our own, more than ready to participate in the risks.

SECTION

XX

LOW EMISSION BURNERS FOR STEAM GENERATION

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B.C.
13 - 14 April, 1988

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Dr. Waibel was Assistant Director of Industrial Energy Utilization at the Institute of Gas Technology, Chicago. He is currently Technical Director, Burner Division of the John Zink Company where his prime responsibility is Research and Development.

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LOW EMISSION BURNERS FOR STEAM GENERATION

INTRODUCTION

It is now commonplace to see restrictions on pollutant emissions as part of the specifications for new boilers and even for retrofit burner applications all over the world. Certain parts of the United States, such as California, have long had restrictions on NO_X emissions and the New Source Performance Standard for Industrial Boilers has extended restrictions to all new boilers in the U.S. greater than 100 million Btu/hr input. It has also become routine, however, to see NO_X specifications for boilers going to Canada, Europe, the Middle East and Asia. West Germany, for example, has enacted exceedingly stringent emissions regulations that effect both existing as well as new combustion equipment.

Although NO_X is the primary pollutant species, other than SO_X, that is regulated, other pollutant species are also becoming the target of increased regulation. These species include carbon monoxide, unburned hydrocarbons and particulates.

The John Zink Company has been a leader in low emission burner development since the first concerns regarding NO_X. Today more than half of the burners it sells are of a low NO_X design. John Zink Company maintains a research and development facility with nine test furnace and boilers that are used to develop and test full scale burners from less than 1 million Btu/hr to over 250 million Btu/hr.

NO_X EMISSIONS

A great deal of progress has been made over the past twenty years in terms of understanding the factors affecting NO_X formation and determining methods for controlling NO_X emissions. Low NO_X burners are available today that reduce NO_X emissions more than 50 to 80% compared with standard burners.

There are two basic sources of NO_X emissions from flames and they are the product of different mechanisms. The resulting NO_X is termed either thermal NO_X or fuel NO_X, depending on the formation process.

FUEL NO_X

Fuel NO_X results from the oxidation of organically bound nitrogen present in the fuel. Fuel nitrogen is typically found in liquid or solid fuels, however, fuel nitrogen can be found in some gaseous fuels in the form of NH_3 or HCN.

Fuel NO_X emissions increase with the concentration of bound nitrogen in the fuel. The fuel nitrogen concentration and excess air level are the most important factors effecting fuel NO_X emissions.

THERMAL NO_X

Thermal NO_X results from the fixation of the N_2 present in the combustion air or fuel. The rate of thermal NO_X formation is sensitive to the excess oxygen and the peak temperatures in the flame zone. Factors that effect the local temperatures in the flame zone have a significant effect on NO_X emissions because the emission rate is exponentially dependent on temperature. Design and operating parameters that influence thermal NO_X emissions include such things as fuel composition, air preheat temperature, excess air, furnace heat release density and heat absorption rate.

Fuel composition is important because the composition directly effects the flame temperature. The adiabatic flame temperature for a given fuel is a good indicator for the potential emissions from that fuel. Inert species, such as carbon dioxide, that reduce the flame temperature reduce the thermal NO_X emissions. Fuel species, such as hydrogen, that produce high flame temperatures result in the high thermal NO_X emissions. This effect is shown in Figure 1. The lower curve shows that increasing carbon dioxide in the fuel results in lower emissions. Fuel from landfills, which contain 50% CO_2, produce NO_X levels that are less than half of those of methane alone. Likewise the upper curve shows that high concentrations of hydrogen, which is often found in the waste gases that are burned in refineries, can result in elevated NO_X emissions.

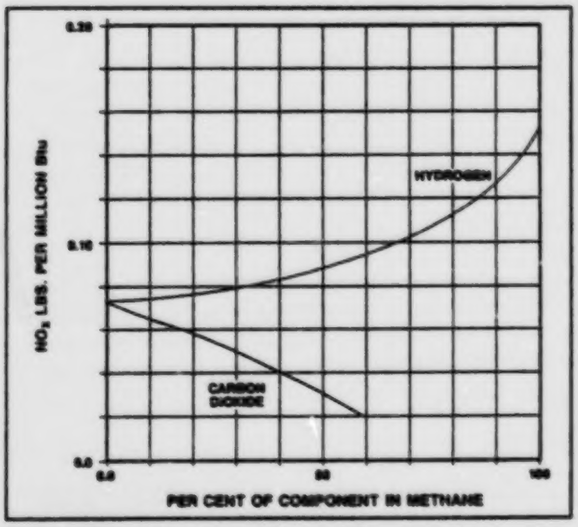

Figure 1. Effect of fuel composition on NO_x emissions: comparison of hydrogen and carbon dioxide

Air preheat temperature effects the flame temperature as well and higher air preheat increases NO_X. Figure 2 shows how the NO_X emissions rise exponentially with air preheat. Preheating combustion air can improve efficiency and result in reduced fuel consumption, however, there is generally a net increase in NO_X emissions because of the exponential dependence of NO_X on flame temperature.

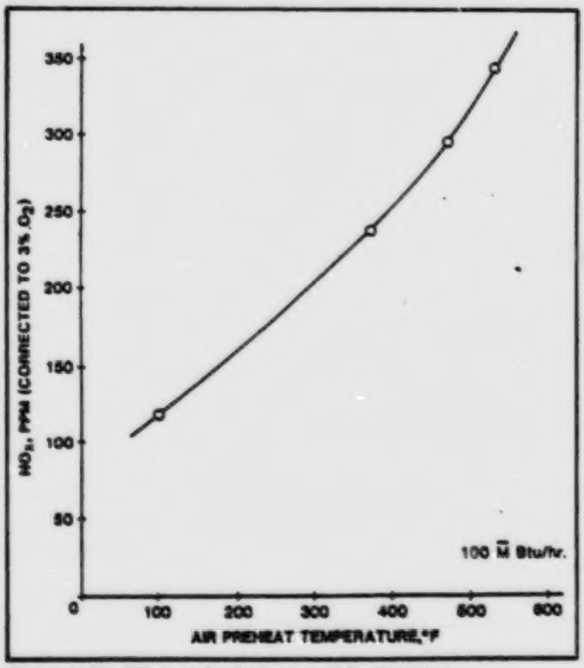

Figure 2. The Effect of Air Preheat on NO_x Emissions

Excess air can increase NO_X because it increases the availability of oxygen atoms in the combustion zone. Low excess air operation is good operating practice since it not only maximizes thermal efficiency, but minimizes thermal NO_X. If the excess air is increased to very high levels it can produce relatively low NO_X levels because of the reduction in flame temperature that results from the high level of excess air. The reduction in flame temperature eventually overcomes the effect due to high O_2 levels. This relationship is shown in Figure 3.

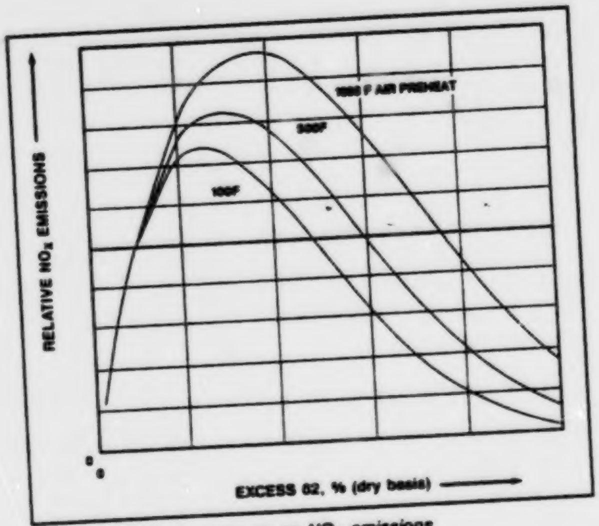

Figure 3. Effect of excess air on NO_x emissions

The furnace heat release density and heat absorption rate are important parameters related to NO_x emissions because they effect the actual temperatures achieved in the flame. If the heat release density in the combustion chamber is high or the heat absorption rate from the flame is low the resultant peak flame temperature will be high and NO_x emissions will be high. Large, conservatively designed boilers tend to produce lower emission levels. The gas temperature leaving the flame zone or at the exit of the radiant combustion chamber is an indication of the potential NO_x emissions when comparing otherwise similar funaces or boilers.

EXPRESSING EMISSION LIMITS

NO_x emission limits can be expressed in a wide variety of ways. For regulatory purposes the NO_x is expressed as NO_2, since it is assumed that it eventually transforms to this species in the environment. Limits are then sometimes specified as a maximum rate of emission of pounds or kilograms of NO_2 per hour, day or year. This method of expressing a limit is relatively unambiguous and requires that optimum emission control be exercised during the period of maximum fuel input. Pounds per million Btu of heat input is another relatively unambiguous method of specifying an emission limit, however, this limit can either be specified at a prescribed heat input rate or it may be required over the entire operating range. Unfortunately, it may be difficult for the operators to maintain low NO_x operating conditions over the entire operating range.

Most European nations have chosen to use NO_x concentration in the dry flue gas as their emission limit. The concentration is typically expressed as PPM (parts per million) or milligrams per normal cubic meter. The concentration is standardized to a specified oxygen concentration by adding or subtracting air. This method of specifying a limit is unfortunate because it does not necessarily apply NO_x limitations uniformly. Depending on the fuels fired, it is possible for one boiler to emit more pounds of NO_x per hour than a second boiler while the first is in compliance and the second not in compliance with the same emission concentration limit. This can happen because, if the concentration limit is expressed in terms of dry flue gas, it penalizes hydrogen rich fuels that produce primarily water as their products of combustion. When the water is removed, the NO_x is concentrated in the remaining dry flue gases. The dry flue gas concentration will be higher than the concentration found for a fuel with less hydrogen, even though the NO_x concentrations were equal in the wet flue gases.

As an example of the disparity that is possible, assume that one boiler is firing natural gas and producing 0.1 lbs. of NO_x per million Btu. The corresponding flue gas NO_x concentration for this fuel would be 84 PPM (dry basis, 3% O_2). Assume that a second boiler is firing a fuel composed of 35% methane and 65% hydrogen and producing only 0.095 lbs. of NO_x per million Btu. The corresponding flue gas concentration for this fuel is 90 PPM. If the prescribed emission rate were set at 85 PPM the first boiler would be compliance and the second would not be in compliance even though the second boiler is emitting less NO_x .

LOW NO_x BURNERS

Early low NO_x technology involved air staging and flue gas recirculation to control NO_x emissions. More recent advances have found fuel staging as a more beneficial method for reducing NO_x emissions.

Air staging requires that the combustion air flow be split into two or more streams. The fuel is burned with part of the air in a fuel rich primary combustion zone and the remaining air is then added downstream of the primary combustion zone to complete combustion of the fuel. NO_x is reduced because a large portion of the heat is released in the primary combustion zone were excess oxygen is not available and thus NO_x formation is not promoted. Thermal NO_x reductions of 30 to 50% can be achieved with this method. Figure 4 shows a schematic view of an air staged burner.

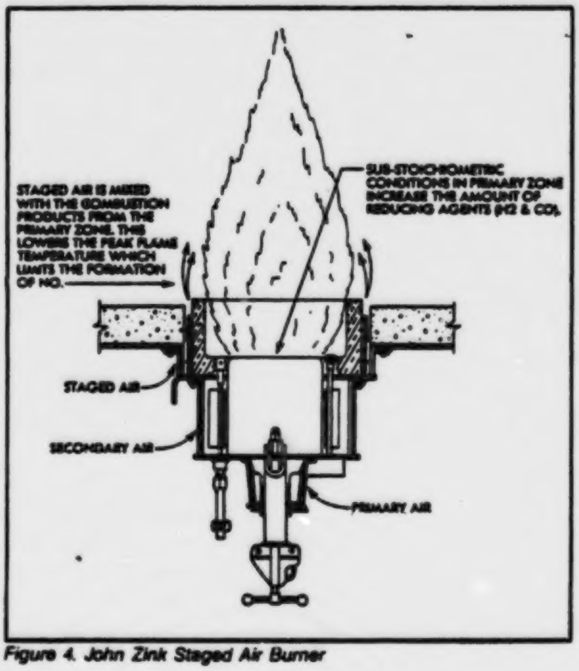
STAGED AIR IS MIXED WITH THE COMBUSTION PRODUCTS FROM THE PRIMARY ZONE. THIS LOWERS THE PEAK FLAME TEMPERATURE WHICH LIMITS THE FORMATION OF NO.
SUB-STOICHIOMETRIC CONDITIONS IN PRIMARY ZONE INCREASE THE AMOUNT OF REDUCING AGENTS (H2 & CO).
STAGED AIR
SECONDARY AIR
PRIMARY AIR

Figure 4. John Zink Staged Air Burner

Air staging is still the most effective method for minimizing fuel NO_x formation. The fuel nitrogen is released in the fuel [illegible] primary combustion zone and tends to form N_2 rather than NO_x because of the scarcity of oxygen atoms. Residence time, temperature and stoichiometry in the primary combustion zone are the factors determining fuel NO_x. To minimize fuel NO_x emissions it is necessary to have high primary zone temperatures, sufficient residence time to allow decomposition of the fuel bound nitrogen and fuel rich stoichiometry.

Flue gas recirculation can be used to reduce thermal NO_x by minimizing the peak temperatures in the flame. Adding flue gas to the combustion air adds inert products of combustion that reduce the adiabatic flame temperature. This reduces thermal NO_x but has little or no effect on fuel NO_x. Although effective, flue gas recirculation has its price in terms of increased complexity in equipment and in operation. Hot, potentially corrosive flue gases are extracted from the stack or furnace and are re-injected into the combustion air or directly into the flame, generally by means of a hot fan. The ratio of flue gas recycle to combustion air must be maintained within acceptable limits over the entire boiler operating range. If the level of flue gas recirculation is too high it will cause flame instability or extinguish the flame. If it is too low the NO_x emissions may exceed the required level. This typically requires that the recycle flow be measured and the flow needs to be carefully controlled within safe and effective limits.

STAGED FUEL LOW NO_x BURNERS

Fuel staging, or reburning as it is sometimes called, was developed by the John Zink Company as a NO_x control technique. The reburning concept was originally utilized to minimize NO_x emission from waste gas streams containing high levels of NH_3 or NO_x. Fuel staging was then applied to burners and has proven to be the most effective means of minimizing NO_x while overcoming the operational problems of air staging and flue gas recirculation.

Figure 5 shows a schematic view of a John Zink staged fuel burner depicting the process. NO_x con-

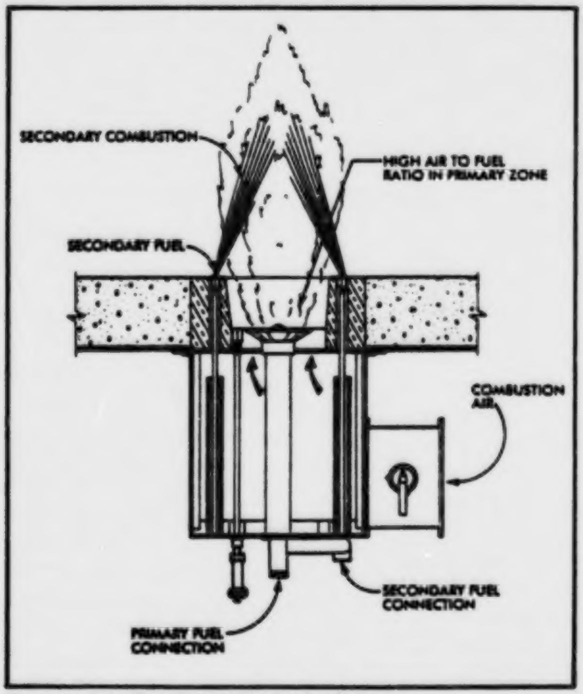
SECONDARY COMBUSTION
HIGH AIR TO FUEL RATIO IN PRIMARY ZONE
SECONDARY FUEL
COMBUSTION AIR
SECONDARY FUEL CONNECTION
PRIMARY FUEL CONNECTION

Figure 5. John Zink Staged Fuel LoNox™ Burner

trol is accomplished by injecting part of the fuel into the bulk of the combustion air to form a lean primary combustion zone and the remainder of the fuel into the tail of the primary zone. Thermal NO_X production in the primary zone is limited by the low flame temperatures that result from the high excess air levels in this lean zone. NO_X production in the primary zone is low, in spite of the high oxygen concentrations, because of the exponential dependence of NO_X formation on the peak flame temperatures. Quenching of the combustion process by the high excess air levels limits the peak flame temperatures and also provides partial combustion products that enhance NO_X reduction in the secondary combustion zone.

NO_X formation in the secondary combustion zone is limited because the air now contains the products of combustion from the primary fuel. These inert combustion products lower the local concentration of oxygen and reduce the peak flame temperatures in the secondary combustion zone. The resulting NO_X levels are substantially lower than those achievable with air staging and are comparable to those found with FGR. Figure 6 shows a comparison of NO_X from a staged fuel boiler burner and a standard boiler burner, each designed and operated at 100 million Btu/hr. The data shows that the fuel staging process provides a significant reduction in NO_X and less sensitivity to air preheat temperature than a conventional burner.

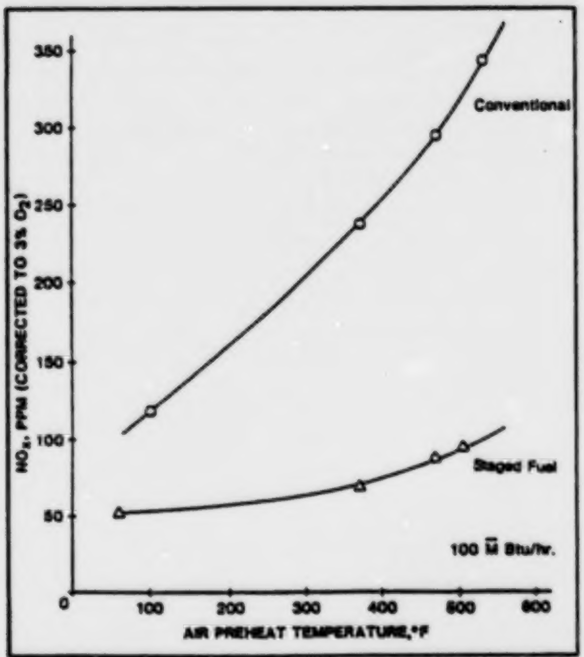

Figure 6. Comparison of NO_X vs. Air Preheat at 2% Excess Oxygen for a Conventional and a Staged Fuel Burner.

Figure 7 shows a cross sectional view of a John Zink AVC staged fuel boiler burner. Since the staged fuel combustion process is lean in both primary and secondary combustion zones, the burner can operate at much lower excess air levels and produces a shorter, crisper flame than a staged air burner with much less potential for forming carbon monoxide, unburned hydrocarbons or particulate. In addition, it can produce NO_X reductions comparable to flue gas recirculation without the costs and potential problems associated with operation of an FGR system.

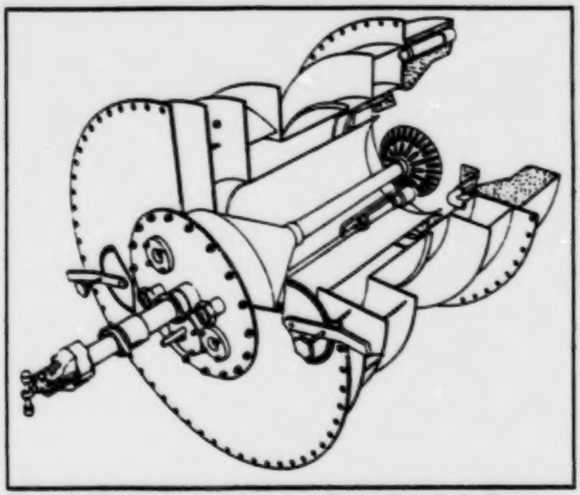

Figure 7. Cutaway View of John Zink Staged Fuel Boiler Burner

If extremely low levels of NO_X are required, however, it is possible to achieve very low emission rates by combining fuel staging with flue gas recirculation. Figure 8 shows data for a staged fuel AVC burner operating with 500 deg F air preheat and various levels of flue gas recirculation. The data shows that the effectiveness of FGR diminishes as the level of FGR is increased until it finally levels off and eventually results in flame stabilization problems. Optimum levels of FGR are

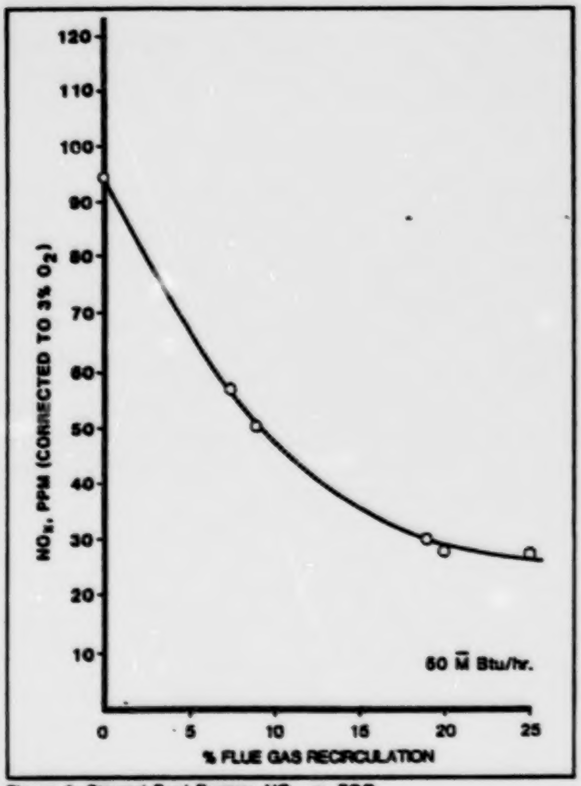

Figure 8. Staged Fuel Burner: NO$_X$ vs. FGR (2% O$_2$, 500°F Air)

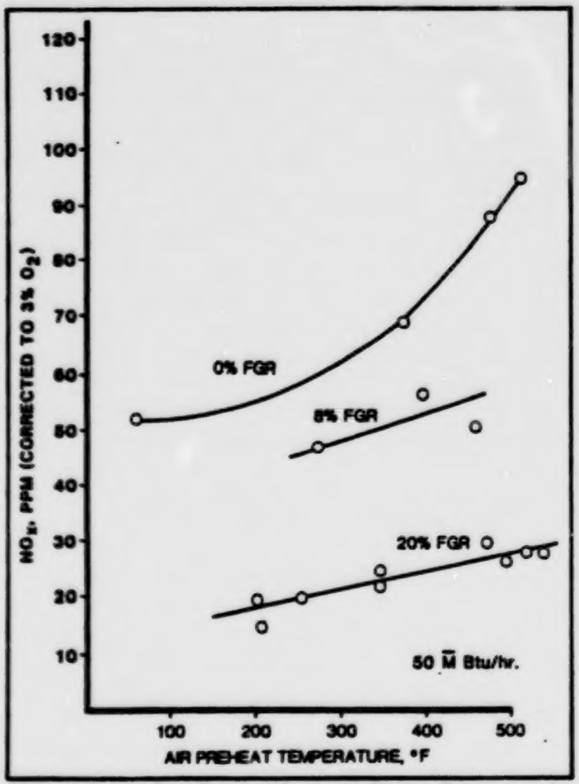

Figure 9. Staged Fuel Burner: NO$_X$ vs. Air Preheat Temperature for Various FGR Rates (2% O$_2$)

typically in the range of 15 to 20%. Figure 9 shows additional data with air preheat levels from ambient to 500 deg F. NO$_X$ emissions from conventional burners would be from 120 to over 300 PPM over this same range. The data, collected with a full scale 50 million Btu/hr burner, show that the combination of FGR and fuel staging can achieve NO$_X$ levels that should meet even the most stringent NO$_X$ requirements.

SUMMARY

A variety of combustion modification and burner design methods are available for minimizing NO$_X$ emissions from boilers. Different methods provide optimum results depending on whether fuel NO$_X$ or thermal NO$_X$ needs to be controlled. Air staging is the most effective burner design technique for controlling fuel NO$_X$, while fuel staging is a more effective design for controlling thermal NO$_X$. Optimum thermal NO$_X$ control can be achieved by combining fuel staging and flue gas recirculation, although at an increase in cost and operational complexity.

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SECTION

XXI

ENERGY MONITORING FOR PLYWOOD VENEER PLANTS

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Richard Green
Neptune Dynamics Ltd.

9/5/666

ABOUT THE SPEAKER:

Mr. J. Richard Green, P. Eng. is President of Neptune Dynamics Limited and is responsible for overall control of the Company. He holds BASc Mechanical Engineering (1966); MASc Mechanical Engineering, Heat Transfer (1970); P. Eng. (1968). In 1975 Mr. Green was awarded "The Award of Excellence in Engineering Design" by the Consulting Engineers of Canada, for his work on fish processing machinery, and later that year he was awarded a Governor General's Medal also for excellence in engineering design. Mr. Green is active with the B. C. Association of Professional Engineers. Mr. Green has over twenty years experience in a wide variety of heat transfer, energy conservation, machinery development and fluid projects.

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PLYWOOD VENEER DRYING PRODUCTION
AND ENERGY MONITORING

Prepared by:

Neptune Dynamics Ltd.

PRODUCTION AND ENERGY MONITORING FOR PLYWOOD VENEER PLANTS

J. Richard Green, P.Eng.
Neptune Dynamics Ltd., Vancouver, B.C.

INTRODUCTION

As part of an effort to improve the energy efficiency of an industrial veneer dryer at the British Columbia Forest Products Limited (B.C.F.P.) plant, research has been performed to determine the effects of various parameters on veneer drying rate and energy consumption. Research applicable to the computerized veneer drying also has been performed to obtain basic information about the multitude of variables that we attempt to control in drying of veneer. The outcome of these investigations has been applied to the #2 veneer dryer at the B.C.F.P. plant, Delta Plywood Division and the results are the subject of this paper.

1. Plant Investigation

1.1 General Procedure

The general procedure followed for investigating dryer performance was to observe the drying operation under the actual production condition. Selected loads of test veneer were fed into the dryer and then segregated at the out feed end of the dryer. Drying time, gas flow rate, number of redry, dryer temperature and pressure were measured during each test. Collected data was later analyzed to construct heat and mass balance for the dryer and to calculate overall performance.

1.2 Veneer Condition

Generally, green veneers are hand sorted in accordance to their moisture content before being fed into the dryers. Depending on the type of wood, moisture content varies as much as 9 - 70% of total volume. The dried veneer ideally has a moisture content of 5 - 6%. The difference in moisture content of the veneer entering and leaving the dryer is the moisture removed from the veneer in the dryer. Using dried veneer generated during the test period, it is possible to calculate the energy of evaporation and the energy required to heat the veneer. This data, plus the gas consumption rate is used to set up heat and mass balance for the dryer.

1.3 Dryer Description

The dryer is made by Moore Dry Kiln company of Oregon. It is a gas-heated, six line, two-zone, longitudinal-flow dryer. The length of the dry zone is about 49 feet, and length of the green zone is about 41 feet, for a total drying length of about 90 feet. A sketch showing general features of the dryer is illustrated in Figure 1. Stacks from the cooling zone are not shown in the sketch.

1.4 Dryer Operation

At the start of the project, the dryer operation was bogged down by numerous fires and an unacceptable percentage of redry veneers. Actual observation of the drying procedure and physical examination of the dryer pin pointed the source of the problem. The responsible parameters were the leaky doors and erratic operation of the dryer. The defective doors allowed the plant temperature air to enter the dryer causing an imbalance of drying temperatures and creating a fire prone atmosphere. The leaky doors were identified and fixed. The erratic operation was the result of frequent stopping of thedryer, leading to a large production loss through over-cooked veneers. The operational problem was reported to the plant manager for corrective action. Upon elimination of operational problems, experimental investigations of production rate, energy consumption, dryer speed, redry rate and other important parameters were carried out.

1.5 Vent Stacks Measurement

Vent stack gas flow rate is measured by taking a velocity transverse with a pitot tube. The stack gas temperature was determined by using thormocouples in thermopile formation. The wet bulb temperature was found by wrapping a cloth around the thermopile rod and then placing it in boiling water. The wet thermopile was placed perpendicular to the stack longitudinal axis, and the wet bulb temperature determined by noting the lowest temperature recorded by the output device.

The stack moisture content was calculated by using the psychrometric equation given in Reference 1:

$$e = e' - PA(t-tw)$$

where e is partial pressure of water vapor at the dry-bulb temperature, t: e' is saturation pressure of water vapor at the wet-bulb temperature, tw: P is the total pressure (essentially atmospheric); A is psychrometric constant

$$A = 3.67 [1 + (tw -32)/157] \times 10^{-4}$$

Upon calculation of the partial vapor pressure (e), the percent water vapor in the stack could be found by the relation:

$$M.C. = (e/p) \times 100$$

Where M.C. is the moisture content, percent by volume, of the stack gas.

2. Instrumentation

The first step to conserve energy in existing industrial dryers is to measure the energy used per production unit and analyze the specific energy consumption with respect to a target figure. Target energy use figures are difficult to obtain for the existing dryer except by direct measurement of the heat transfer/energy usage under varying conditions. Of course, the dryer production is a function of many interrelated parameters and to minimize energy usage, while still maintaining adequate production, is to continuously measure the important variables for processing and then adjust the dryer operation for optimum performance. The important parameters and the required instrumentation are discussed below.

2.1 Gas Consumption

To set up an energy balance knowledge of accurate gas consumption is required. Accurate gas meters (Rockwell G610201-579) were selected by NDL to measure output which was processed and fed into a computer to print out the dryer's gas consumption at the end of each shift. The gas flow rate could also be monitored at any desired time.

2.2 Water Flow

In normal operation, no water should be sprayed in the dryer, however, it was discovered that during some shifts, cold water was piped throughout the drying zone as a means to extinguish and prevent dryer fire. As the energy efficiency of the dryer was greatly reduced by the presence of unwanted cooling water, it was decided to stop this practice by controlling the water flow. Various means of control such as pressure switch, flow sensing element and flow meters could be used. However, use of pressure switch and flow sensing elements was limited by fluctuation of line pressure. Thus, a flow meter with electrical output signal was needed. A paddle type water meter, Signet Scientific Paddle Wheel Flosensors, Model MK 515, was selected for this purpose. As soon as the water tap opened and water started to flow into the dryer, a signal is sent out by the sensing flow meter. This signal is processed and fed into the computer which records the exact time and duration of water flow into the dryer. This enables management to pin point the operational difficulty and take corrective actions.

2.3 Dryer Speed

The instrument utilizes magnetic pickups as pulse generators which are mounted to sense passing teeth on a feed gear and send pulses to a signal conditioning instrument. This type of monitoring gives an exact account of dryer feed rate regardless of the drive motor or the gear ratios in the gear box. The output of the sensor is processed and fed into the computer for display in the operators monitor screen and is recorded for further evaluation of the dryer performance.

2.4 Veneer Counter

To obtain a clear and accurate picture of veneer production rate, it is desirable to count the number of dried veneer. A special counting instrument was designed and built by the NDL staff which uses Photo switches for sensing veneer movement and a custom built signal conditioner to process the output which is then fed into the computer for further use. An important aspect of this counter is its capability to count 1/2 or 1/4 sheets so the associated software can calculate the exact production rate with a 98% degree of accuracy.

2.5 Redry Counter

To prevent over drying of the veneer, and to increase production, it is desirable to operate the dryer so as to produce 10 - 15% redry. To implement and achieve this goal a redry counter is necessary. Careful examination of the existing moisture sensor at the dryer outlet revealed their possible use as a redry counter as well. Auxiliary electronics were designed and installed by NDL resulting in a reliable redry counter. The output of the redry counter is also processed and fed into the computer for display in the operator's monitor screen and is recorded for further evaluation of the dryer performance.

2.6 Moisture Content Inside the Dryer

A crucial parameter in monitoring dryer efficiency and capability, is the moisture content (MC) inside the dryer. Over dryed product, i.e. low MC, means slow dryer speed and/or high drying temperature. Reversely, high MC indicates high dryer speed and/or low drying temperature. Thus, the correct percent of MC is a must for optimum operation of the dryer.

Due to high gas temperature, standard humidity meters cannot be utilized here. Consequently, a suitable instrument was designed to do the job utilizing RTD probes along with appropriate signal conditioning devices to process and record the wet and dry bulb temperature from which MC was calculated.

2.7 Dryer internal temperature

Recording of the dryer internal temperature at different locations will assist the supervisory personnel to determine if the dryers are being properly operated. Furthermore, it is a good way to locate "Hot" and "Cold" sections in the dryer, particularly, the air temperature at the inlet and outlet of each zone. The temperature difference between the air entering and air leaving is the temperature drop through the drying area. This temperature drop is an indication of the amount of work done by the heated air in evaporating water from the veneer in the drying area. To measure and record this temperature, type "J" thermocouples were used. Location of the thermocouples is indicated in Fig. 1.

3. Data Processing System

Over the last few years, there has been a great trend toward utilization of personal computers (PC's) for industrial application by control engineers, Reference 3. These computers are readily available, moderately priced and above all can use many available software. NDL selected and used an IBM PC for the data processing task, and proceeded to develop special software and select appropriate instrumentation for the proposed control system. Figure 13 shows a schematic drawing of the NDL computerized data collection and control system for the BCFP veneer dryers. The designed system was put into operation as of June 1986 and gradually debugged. The installed computer plus the developed hardware has enabled us to collect, analyze and record an enormous amount of data leading to the discovery of important factors responsible for efficient dryer operation. Examples of such data are continuous recording of gas, water and veneer flow into the dryer. Data has been collected for the second part of 1986 and all operational periods in 1987.

3.1 Display and Printing

Only important and controllable parameters such as dryer speed, internal temperature, veneer count, and the number of redrys will be displayed on the operator's screen. This data will assist the operator to have a continuous and clear picture of the dryer's performance during his shift. Other related data will be processed and stored by the computer. A print out of dryer's performance such as total gas consumption, percent of operation time, percent of water on/off period and other desirable information will be printed out by an in house printer. A sample of this print out is shown in Table 1.

4. Results

The results of the tests are given in Table 2. For the sake of comparison, results of previous investigations are included as well. Calculated results for heat input and distribution are summarized in Table 3.

It should be pointed out that the previous results taken from Reference (2) are for the time when large quantities of water were poured inside the dryer to curtail dryer fires.. This explains why a great portion of heat input is spent to evaporate water from the bottom of the dryer causing a great energy penalty for such an operation.

Table 3 gives a sample computer recording of dryer's off-on period during each shift. It became clear that, during some shifts a great portion of the shift (60%) the dryer is stopped. This type of operation not only brings down the dryer production but it is also responsible for over-dried (cooked) veneer. To obtain any meaningful result, this type of operation must be corrected.

Figure 3 is a schematic sketch showing hourly flow rates of material (except gas) into and out of the dryer #2 before the start of this project. A similar representation for one of the tests conducted on June 1986 is shown in Figure 4. The two figures are quite similar with the exception of the excess water flow rate. During the latter test, care was taken to prevent dryer fire and thus avoid excess water flow through the sprinkler system. In both cases, the amount of water removed from the veneer is about 2.2 tons/hr. However, with the large amount of water poured inside the dryer as means of fire prevention in the first case, a great energy penalty, as large as 47.4% has to be paid. While, in the latter case, this penalty is reduced to 16.3%. Of course, this is not desirable and should be totally eliminated. Figure 5 and 6 show material flow rates into and out of the dryer #2 during some tests on April 1987. Here, the amount of sprinkled water into the dryer is minute. However, the difference is between the percent of the dryer's running time during each shift. It is clear that higher energy efficiency is the immediate advantage of continuous operation. Furthermore, there is another hidden benefit, less over-dryed veneer, which is not shown here.

Results of heat balance for the previous tests are shown in Figure 7. For the sake of quick comparison, the percentage of heat consumption for the dryer is indicated by pie charts in Figure 8. Once again, the detrimental effect of excess water on overall dryer efficiency is quite apparent.

Based on the collected data and actual operation of the dryer, it is quite evident that the first step toward improving dryer's efficiency is to run the dryer continuously as long as possible during each shift. To do so, it is essential to pinpoint the parameters responsible for erratic operation of the dryer. To this end, we instrumented the dryer to the full extent and analyzed the result via a special computer and associated software. Enormous amount of data (15 Mega bytes/week) was collected, however, only the most important results are discussed here.

Dryers as well as press production for the period 1985-87 is presented in Figure 9. As indicated, the average press production continually improves from its lowest at 1985 to its highest at 1987. An important factor for this increase is reduced manufacturing loss as shown in the bottom of the graphs. Another factor responsible for the increased production is continuous operation of the dryer.

Improvement in gas consumption per dryer production is shown in Fig.10. It is worth mentioning that the improvement has been continuous since the completion of the monitoring system in September '87, ranging from 4-12%. This improvement has a cash value of at least $50,000 per year. Further improvement can be achieved through regulating operating temperature and the vent damper settings.

5. Conclusion

Based on the actual observation of the dryer and a huge sum of collected data, the most detrimental factors for the dryer's performance are the stop-start operation of the dryer and the dumping of cold water into dryer via fire fighting sprinklers. Stopping the dryer not only wastes energy but also over-cooks the veneer. Flooding of the dryer with cold water exerts a great energy penalty; as much as 45% in some instances, leading to uneconomical operation. Other parameters responsible for high energy consumption are high dryer temperature and a low percent of redry. Obviously, no energy saving can be realized without first eliminating the above mentioned problems.

To achieve high performance through computerized monitoring and control, we must first have a smooth running dryer. We can then fine tune the dryer by measuring important parameters, process them and thus control the process through the developed software. So far we have been successful in achieving smooth operation through computerized monitored control of the veneer dryer. The complete computerized control is the final objective.

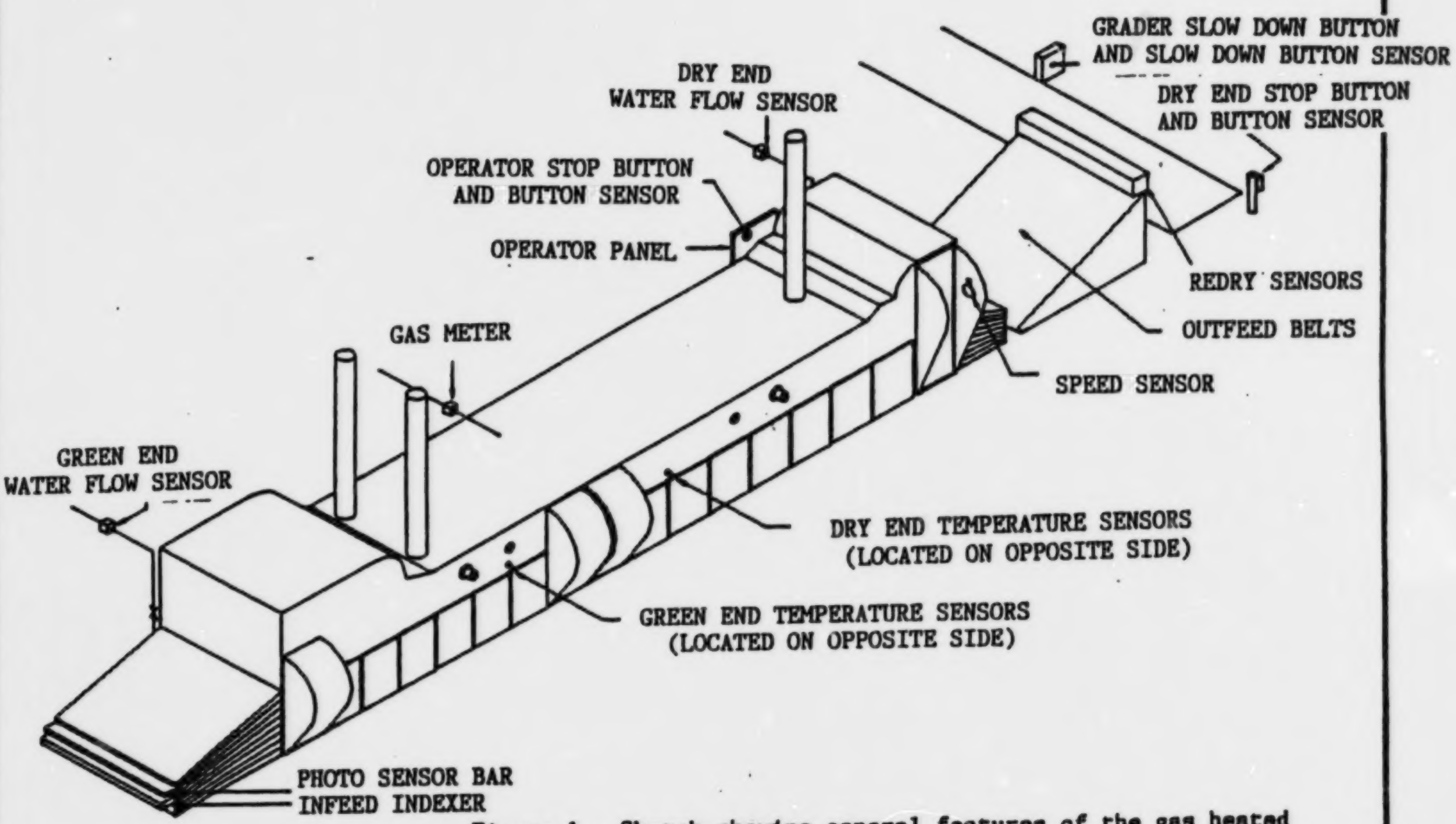
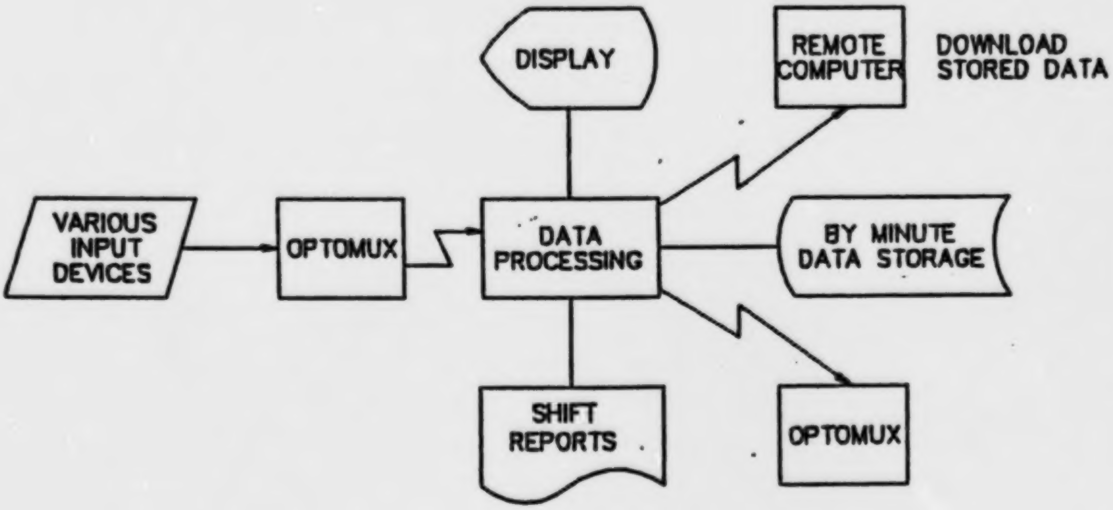

GRADER SLOW DOWN BUTTON
AND SLOW DOWN BUTTON SENSOR
DRY END STOP BUTTON
AND BUTTON SENSOR
DRY END
WATER FLOW SENSOR
OPERATOR STOP BUTTON
AND BUTTON SENSOR
OPERATOR PANEL
REDRY SENSORS
OUTFEED BELTS
GAS METER
SPEED SENSOR
GREEN END
WATER FLOW SENSOR
DRY END TEMPERATURE SENSORS
(LOCATED ON OPPOSITE SIDE)
GREEN END TEMPERATURE SENSORS
(LOCATED ON OPPOSITE SIDE)
PHOTO SENSOR BAR
INFEED INDEXER

Figure 1 - Sketch showing general features of the gas heated two-zone longitudinal-flow veneer dryer and associated instruments



DISPLAY
REMOTE COMPUTER
DOWNLOAD STORED DATA
VARIOUS INPUT DEVICES
OPTOMUX
DATA PROCESSING
BY MINUTE DATA STORAGE
SHIFT REPORTS
OPTOMUX

Figure 2 - Sketch of computerized data collection and processing system for B.C.F.P. veneer dryer

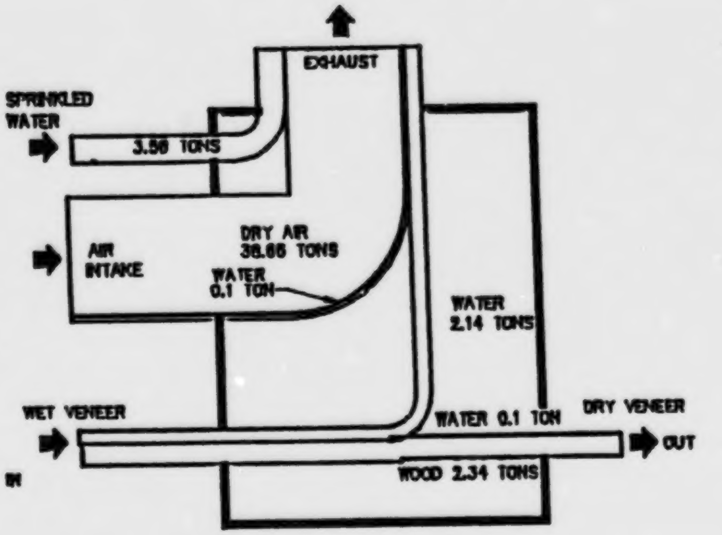

EXHAUST
SPRINKLED WATER
3.56 TONS
AIR INTAKE
DRY AIR 38.66 TONS
WATER 0.1 TON
WATER 2.14 TONS
WET VENEER
IN
WATER 0.1 TON
DRY VENEER
OUT
WOOD 2.34 TONS

FIGURE 3 – A SCHEMATIC FLOW SKETCH SHOWING HOURLY FLOW RATES INTO AND OUT OF THE VENEER DRYER FOR TEST 1.

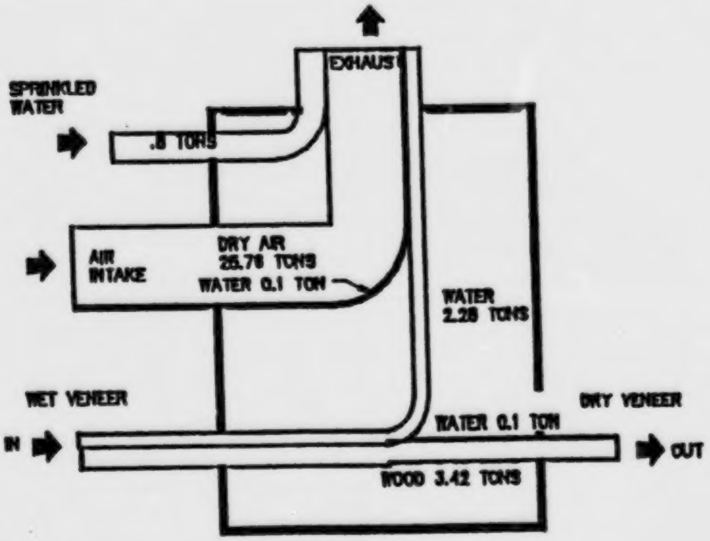

EXHAUST
SPRINKLED WATER
.8 TONS
AIR INTAKE
DRY AIR 25.78 TONS
WATER 0.1 TON
WATER 2.28 TONS
WET VENEER
IN
WATER 0.1 TON
DRY VENEER
OUT
WOOD 3.42 TONS

FIGURE 4 – A SCHEMATIC FLOW SKETCH SHOWING HOURLY FLOW RATES INTO AND OUT OF THE VENEER DRYER FOR TEST 2.

HOURLY FLOW RATES INTO AND OUT OF VENEER DRYER NO. 2 FOR VARIOUS TESTS

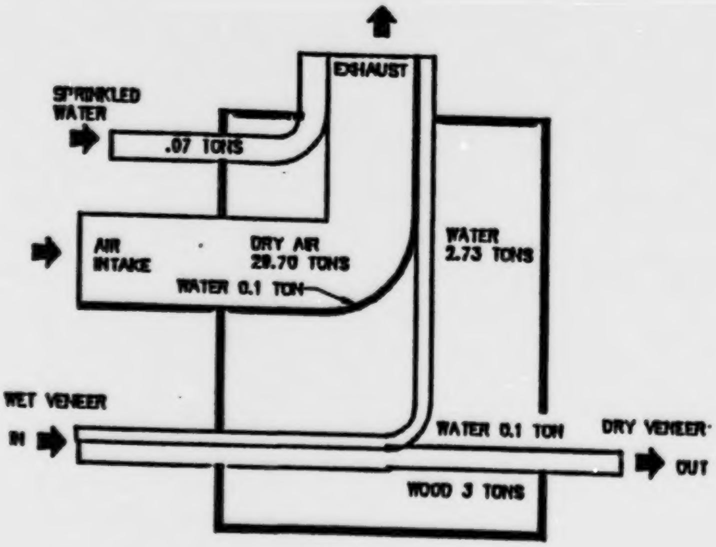

EXHAUST
SPRINKLED WATER
.07 TONS
AIR INTAKE
DRY AIR 29.70 TONS
WATER 0.1 TON
WATER 2.73 TONS
WET VENEER
IN
WATER 0.1 TON
DRY VENEER
OUT
WOOD 3 TONS

FIGURE 5 - A SCHEMATIC FLOW SKETCH SHOWING HOURLY FLOW RATES INTO AND OUT OF THE VENEER DRYER FOR TEST 3.

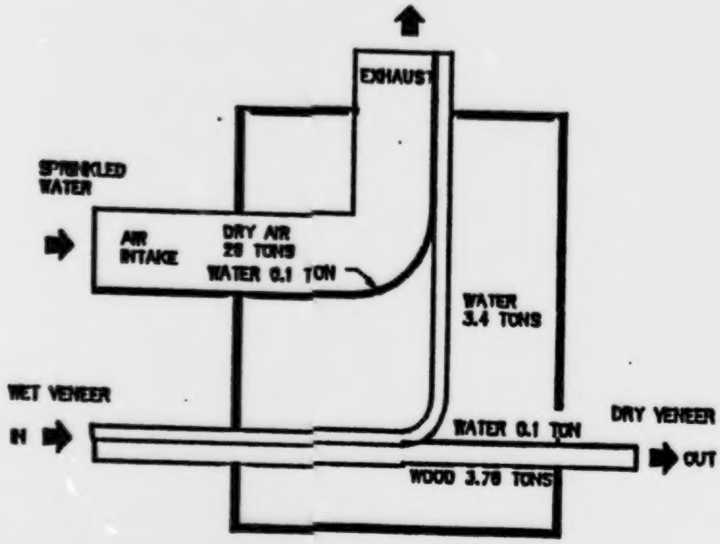

EXHAUST
SPRINKLED WATER
AIR INTAKE
DRY AIR 28 TONS
WATER 0.1 TON
WATER 3.4 TONS
WET VENEER
IN
WATER 0.1 TON
DRY VENEER
OUT
WOOD 3.78 TONS

FIGURE 6 - A SCHEMATIC FLOW SKETCH SHOWING HOURLY FLOW RATES INTO AND OUT OF THE VENEER DRYER FOR TEST 4.

HOURLY FLOW RATES INTO AND OUT OF VENEER DRYER NO. 2 FOR VARIOUS TESTS

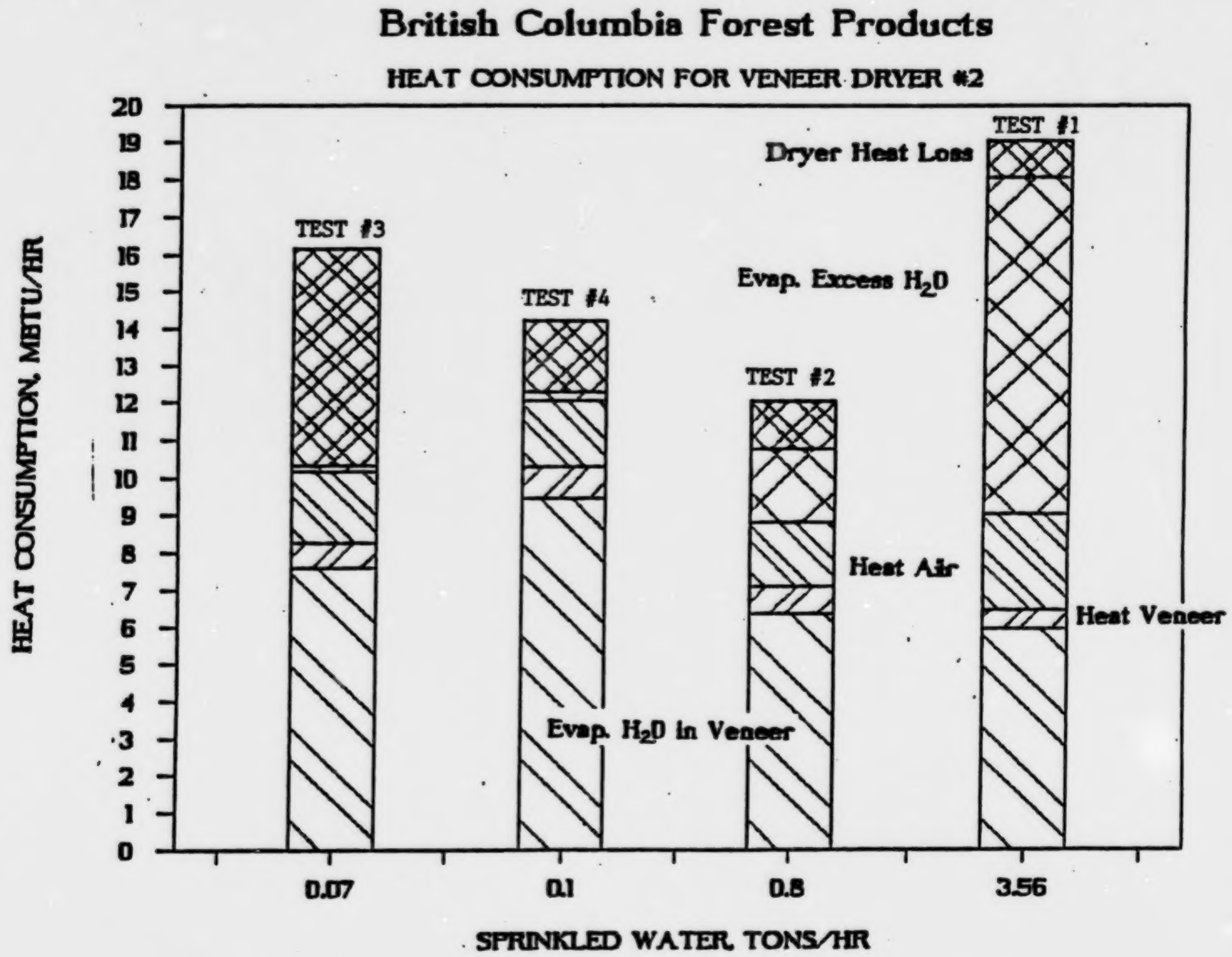


Figure 7. Heat Consumption for Veneer Dryer #2 at Various Test Conditions

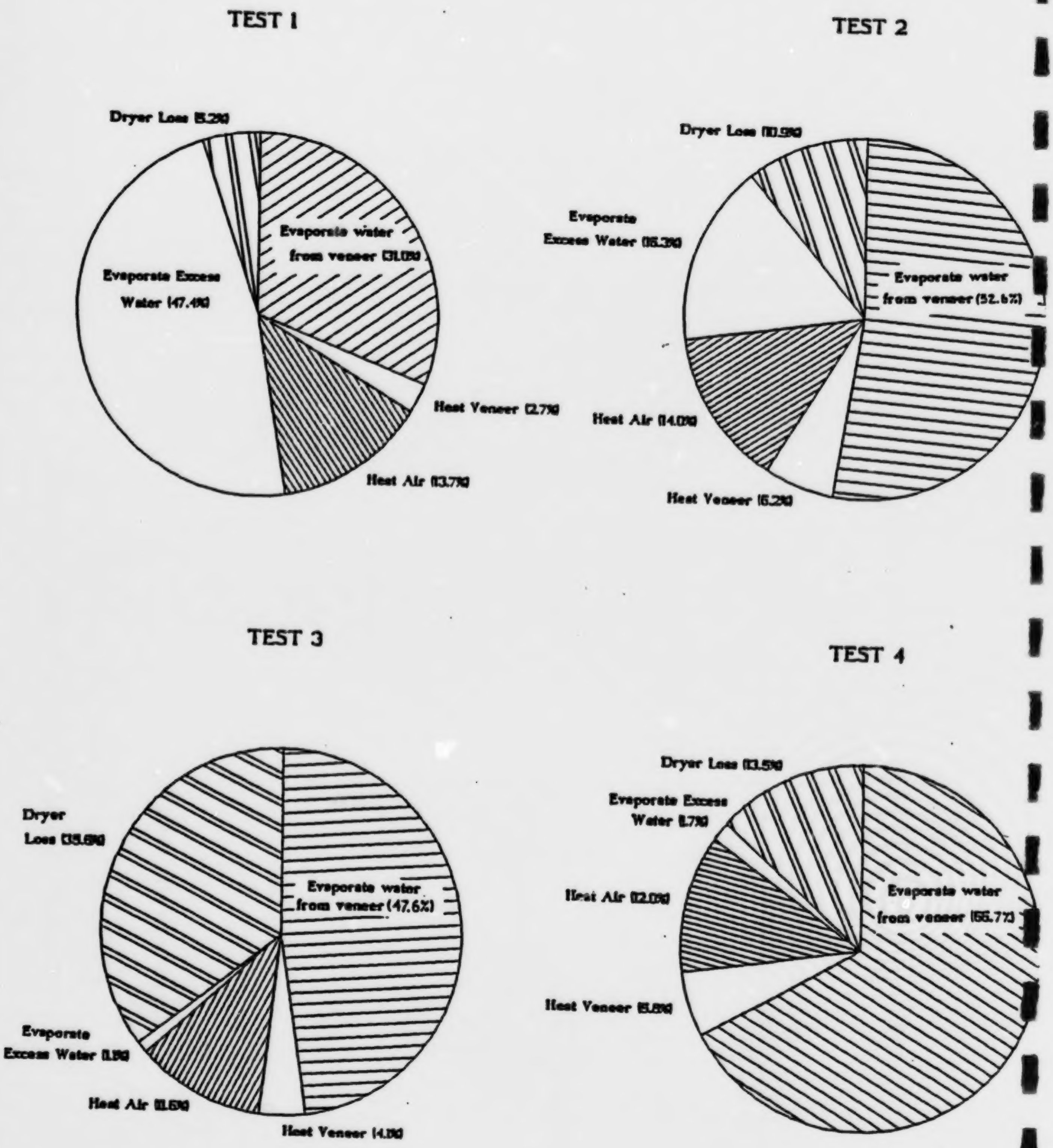


Figure 8. Distribution of Heat Input to Veneer Dryer #2 for Various Tests

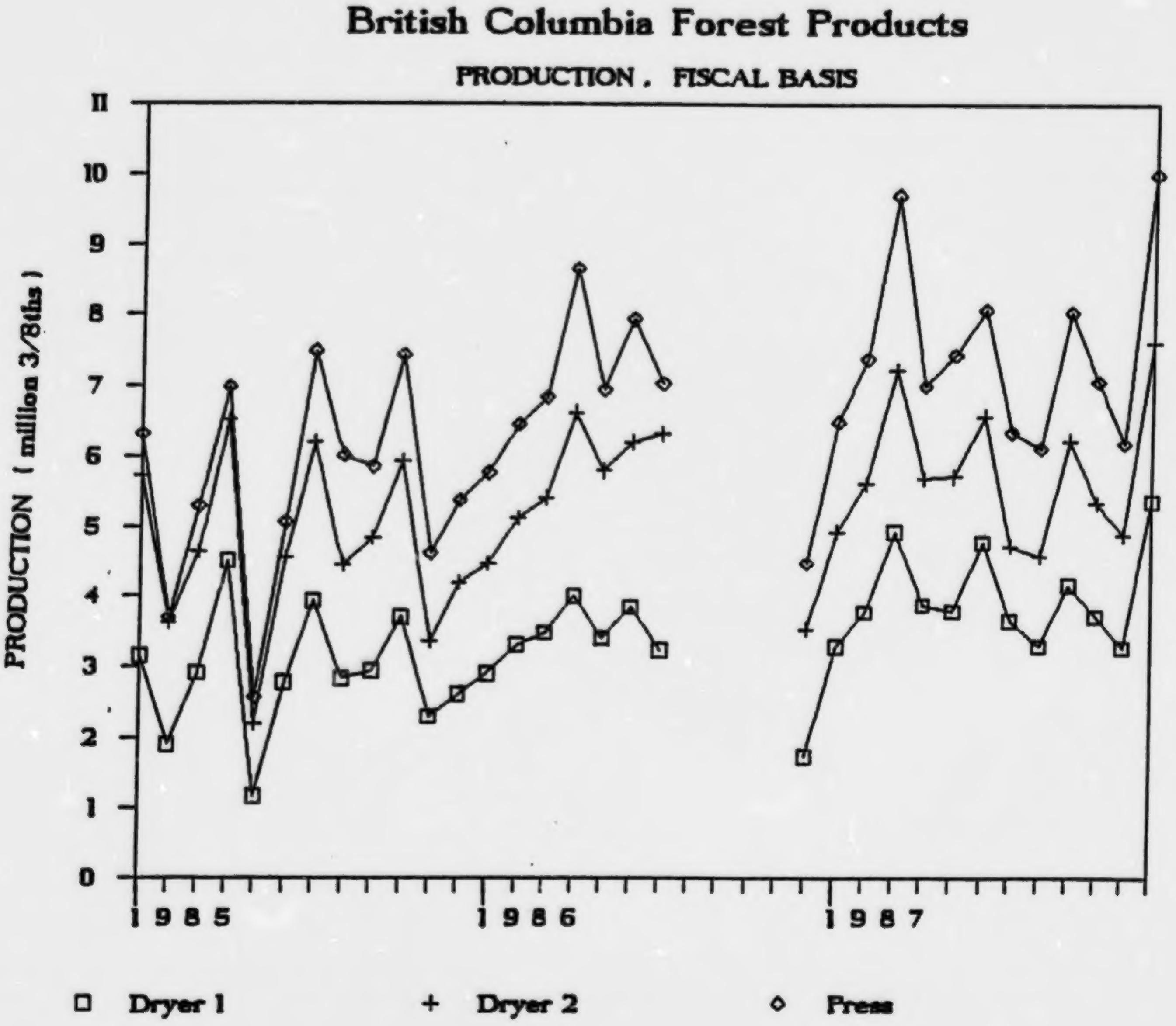


FIGURE 9 - Press and Dryers Production

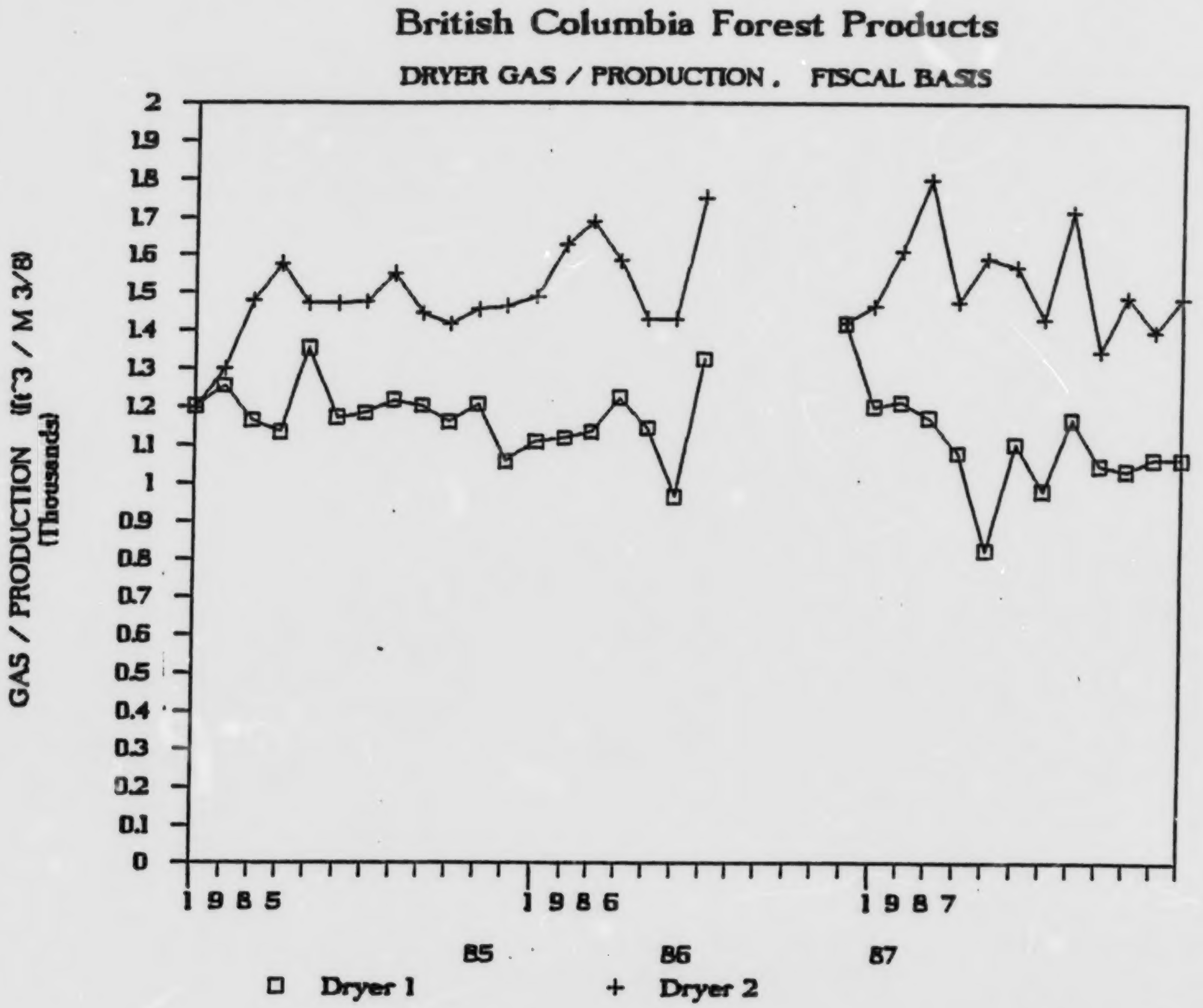


Figure 10. Gas Consumption Improvement for Dryers #1 and #2

```
=================================================================================
==============  B.C.F.P  Plywood Dryer -- Shift Report (Ver 2.08)  ============
=================================================================================

  Date :  Jul 17, 1987                  Total shift length =  8:30:00
 Shift :    Afternoon                Total recording time =  8:30:00 =100 %

                         |           T I M E             |VOLUME |    #
 Dryer # 1               |   ON    |   OFF   |    %     |  GAL  | PRESSED
 ---------------------------------------------------------------------------
     Water          :    | 0:45:18| 7:44:42|    9 %    |  1650 |
                         |         |         |           |       |
     Operation      :    | 6:37:18| 1:52:42|   78 %    |       |
                         |         |         |           |       |
     Stopped        :    |         |         |           |       |
        OUT FEED         |         | 0:43:09|           |       |
        IN  FEED         |         | 0:50:21|           |       |    99
        OPERATOR         |         | 0:19:03|           |       |     4
                         ---------------------------------------------------
     Sub Total      :    |         | 1:52:33|           |       |
     Slowed         :    |         |         |           |       |

   Gas Consumption :        57300 cu.ft.     =  59592,000 BTU
      Average Rate :         6741 cu.ft./hr =    7011,000 BTU/hr

                         |           T I M E             |VOLUME |    #
 Dryer # 2               |   ON    |   OFF   |    %     |  GAL  | PRESSED
 ---------------------------------------------------------------------------
     Water (Green)  :    | 6:34:12| 1:55:48|   77 %    |   250 |
     Water (Dry)    :    | 0:20:30| 8:09:30|    4 %    |   724 |
                         |         |         |           |       |
     Operation      :    | 6:54:12| 1:35:48|   81 %    |       |
                         |         |         |           |       |
     Stopped        :    |         |         |           |       |
        OUT FEED         |         | 0:52:57|           |       |
        IN  FEED         |         | 0:42:42|           |       |    27
        OPERATOR         |         | 0:00:00|           |       |     0
                         ---------------------------------------------------
     Sub Total      :    |         | 1:35:39|           |       |
     Slowed         :    |         |         |           |       |    33

     Sheet Count    :      6725
     Redry Count    :      1453
   Gas Consumption :      77100 cu.ft.     =  80184,000 BTU
      Average Rate :        9071 cu.ft./hr =    9433,000 BTU/hr

=================================================================================
```

Table 1. A Sample of Computer Output for the Shift Report

TABLE 2

TEST DATA SUMMARY FOR THE VENEER DRYER #2

Test Date: Test Number:	30/1/84 (1)	25/6/86 (2)	1/4/87 (3)	2/4/87 (4)
Green Veneer rate, tons/hr	4.48	5.70	5.80	7.16
Dry Veneer rate, tons/hr	2.34	3.42	3.07	3.76
Veneer moisture removal, tons/hr	2.14	2.28	2.73	3.40
Stack Flows:				
Air volume flow, SCFM	19,280	12,880	14,850	13,005
Air mass flow rate, tons/hr	38.65	25.76	29.70	26.01
Water flow rate, tons/hr	5.70	3.08	2.80	3.50
Gas Flow rate, MBTU/hr	19.00	12.07	16.37	14.20
Heat consumption, Btu/lb water evaporated	4,036	2,406	2,726	1,898

TABLE 3

SUMMARY OF HEAT INPUT AND HEAT CONSUMPTION DISTRIBUTION FOR THE VENEER DRYER #2

TEST #	HEAT From the gas	To evap. veneer moisture	To raise temp of veneer	To raise temp of air intake	To evap. excess H O	Dry. Heat loss by difference
			Millions of BTU/hr			
1	19.00 100.0%	5.96 31.0%	0.51 2.7%	2.60 13.7%	9.00 47.4%	0.99 5.2%
2	12.07 100.0%	6.35 52.6%	0.75 6.2%	1.70 14.0%	1.96 16.26%	1.30 10.94%
3	16.37 100.0%	7.60 47.6%	0.67 4.12%	1.90 11.60%	0.17 1.05%	5.83 35.60%
4	14.20 100.0%	9.47 66.7%	0.83 5.82%	1.73 12.0%	0.25 1.73%	1.92 13.50%

REFERENCES

1. Bindon, H.H., 1963. "A Critical Review of Tables and Charts Used in Psychrometry" in: Humidity and Moisture Measurement and Control in Science and Industry, Volume One, Robert E. Raskin (ed), Reinhold Publishing Corporation, pp. 3-15.

2. J.R. Green, A. Sykanda, and J. Renwick, "A Program Developed by an Energy Conservation Study for British Columbia Forest Products," (1984), pp. 9-10.

3. E.J. Compass, "Industrializing the Personal Computer," Control Engineering 32(8) (1985): 70-71.

SECTION

XXII

DEVELOPING THE MARKET FOR NATURAL GAS VEHICLES

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B.C.
13 - 14 April, 1988

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Jeff received his B.A. in International Affairs and Communications from the George Washington University. He obtained his M.Sc. and Ph.D. from the University of London (London, England) in International Relations where he specialized in studies on decision making and conflict resolution studies. He did his dissertation on the effects of the press on foreign policy decision making.

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DEVELOPING THE MARKET FOR NATURAL GAS VEHICLES

THE CURRENT ENVIRONMENT FOR MARKET DEVELOPMENT

Natural gas vehicles have been touted and marketed in the U.S. for the past twenty years under the guise of four principle benefits:

- Favorable economics -- the difference in price between an equivalent gallon of natural gas and a significantly more costly gallon of gasoline;
- Clean burning qualities -- the fact that emissions from engines burning natural gas are generally lower than emissions from gasoline engines, and that less carbon build-up results within engines burning natural gas;
- Safety -- the structural integrity of the on-board fuel system is superior to gasoline systems, and natural gas does not puddle as do liquid fuels when a leak occurs.
- Favorable supply and potential deliverability -- ample in-ground supplies of this North American fuel delivered throughout a million mile pipeline network can help reduce U.S. dependence on imported sources of petroleum.

Fuel Price Economics and the Market Pull

Up until the end of 1985, favorable economics was the driving force motivating conversions to natural gas. With the precipitous drop in petroleum prices (a gallon of gasoline dropped from $1.20 in December 1985 to just under 83¢ one year later[1]) the motivation to convert to natural gas dwindled. This "pull" of economic market forces that existed in days of high oil prices, when an equivalent gallon of natural gas was about 50% less than gasoline, made marketing natural gas vehicles (NGV) conversions much easier. A.G.A. estimates that the previous annual growth rate of conversions at about 10% leveled off by 1987 to about zero, considering new conversions less fleets that removed NGVs from service without converting new vehicles[2].

By the end of 1987, however, the lack of market "pull" of low gasoline prices was eclipsed by other environmental and legislative factors that today (1988) are creating an improved market situation for natural gas and all the alternative fuels.

Environmental Considerations, Oil Imports, and The Market Push

Federal government clean air legislation and environmental regulations have today created a market "push" for natural gas vehicles and other alternative, so-called "clean burning" fuels. Federal initiatives, in turn, have motivated state legislators to devise transportation - based alternative fuel strategies that forbode new and exciting market initiatives for NGVs and other fuel alternatives.

The Clean Air Act[3] has had a major impact on states' actions to develop pollution control policies and strategies. Seventy to eighty major metropolitan communities across the U.S., comprising some 80% of the population, are categorized by the Environmental Protection Agency as non-attainment areas for ozone, nitrous, oxide (NO_x) or carbon monoxide (CO).

Emission standards established by the Environmental Protection Agency (EPA) for diesel engines, regulating particulates and NO_x are forcing engine manufacturers to look at a number of "clean-up" strategies. The most prevalent options being explored include particulate traps, computer controlled fueling and ignition systems, improved engine cooling techniques and -- alternate fuels. In-line exhaust clean up using traps has been a popular option to explore, but the technology as of 1988 is expensive and ineffective as a long-term emission control strategy to meet 1991 standards. The current generation of traps are $1,000-$2,000 and last for less than one year before needing replacement.

The specter of growing U.S. reliance on imported oil is yet another factor motivating alternate fuel commercialization, but is a less visible factor than environmental considerations. According to estimates by the U.S. DOE, the U.S. imports about 43% of its petroleum. According to the DOE, the U.S. will be 50% to 60% reliant on foreign sources of oil by the mid 1990s[4]. Given the fact that approximately 63% of petroleum in the U.S. is consumed in the transportation sector, any sudden disruption in petroleum supply could trigger substantial dislocation throughout the domestic refinery, delivery, and consumption network. Increased use of natural gas in transportation, on the other hand, affords an opportunity to insulate that sector from such dislocation[5].

These combined factors present a "window of opportunity" for alternate fuels, and specifically for natural gas. Federal and state legislation, mostly in response to mandates under the Clean Air Act, should create an increasingly attractive market for the various fuel alternatives.

- Federal legislation (S-1518, Methonal and Alternative Fuels Promotion Act of 1987, HR-3355, The Alternative Fuels Act of 1987) proposes to support fuel alternative initiatives for methanol and natural gas in the form of corporate average fuel economy (CAFE) credits to our manufacturers who produce vehicles burning non-petroleum fuels. Funding for a variety of demonstration programs also is being advocated to support alternate fuels programs.

- State legislation and regulations are creating new market opportunities for alcohol and gaseous fuels. Colorado has mandated alcohol blends up to 10% during January through March to reduce carbon monoxide (CO) emissions. Arizona too, is legislating alcohol fuel blends to reduce CO, and the city of Tucson will require metropolitan bus purchases after 1988 to be alternate fuel equipment. The South Coast Air Quality Management District, regulating fleets in four southern California counties, will require companies with 15 vehicles or more to purchase alternate-fueled vehicles. Half of new purchases in 1993 and all new vehicles purchased after 1994 will be required to be alternate fueled.

NGV EQUIPMENT FOR DEVELOPING MARKETS

Up until 1988 the NGV equipment for gasoline -- and for emerging diesel technology -- had been dominated by mechanically-based retrofit equipment. Vendors of NGV equipment have, for the most part, been small, undercapitalized entrepreneurs who have struggled in the marketplace to perform R&D, manage their respective businesses, and market their product and services. Today a majority of NGV conversions in the U.S. are being done by about half a dozen vendors who have survived the downturn of gasoline prices. The vendors of both gasoline and diesel conversion equipment rely on components manufactured by another two dozen companies, with regulator and carburetion equipment manufactured by about six to eight companies in the U.S. and Europe[6].

Research and development activities often are the precursers to new and continued markets for technology. A number of major developments currently under way will affect the future of natural gas vehicle markets and market development.

Computer Compatible NGV Technology

The handful of NGV vendors and conversion companies are now struggling to develop underhood equipment compatible with the newest generation of computer controlled vehicles being manufactured by the major U.S. and foreign auto companies. CNG Fuel Systems of Canada, which closed its doors as of December 1987, was one of only two NGV vendors developing a computer compatible conversion kit. Air Sensors, Inc., of Seattle, WA, is developing a computer compatible conversion system, but most of the current generation of mechanical equipment is being supplemented by electronic "fixes", instruments that can deactivate or "avoid" computerized vehicle componentry -- oxygen sensors and computer mapping systems -- when the (dual fuel) vehicles are in the natural gas mode. The electronic "fixes" are being employed with varied degrees of success by NGV converters and it appears that retrofit conversions will be able to be compatible with at least some of the newest of engine technologies.

Environmental benefits from these conversions, however, are not likely to offer the same advantages of the relatively clean burning gasoline vehicles produced in 1987 and beyond. As EPA emission standards have tightened, manufactured vehicles have become cleaner burning. It is likely that environmental benefits of converting older vehicles, however, -- there are some 125 million pre-1987 light duty vehicles on the road -- will still play a factor in NGV retrofits. Emissions testing programs are now underway at EPA in Ann Arbor, and will be done in Texas and Arizona to determine in a quantifiable fashion specific benefits.

New generations of gasoline vehicles that are computer controlled will force NGV technologists to improve and refine conversion kits if the market in the future is to survive. While some NGV retrofit specialists may be able to adapt their equipment to fit new vehicles, much of their current technology may be only appropriate for a shrinking population of older vehicles (pre-1987). It is likely that computer controlled NGV technology on new vehicles will come only when a major manufacturers become involved in building an NGV at the factory.

The Role of Original Equipment Manufacturers

Involvement of original equipment manufacturers (OEMs) will be the key to the future success of NGVs in the marketplace, be it for gasoline or diesel vehicles. The automotive 'majors' -- Ford and GM -- indicate tentative willingness to develop dedicated (natural gas only) NGVs if they can be promised substantial demand for NGVs, in the 1,000 to 5,000 per year range with assurances of growth up to and beyond a five year period.[7] Critical to this success will be whether or not the natural gas utility industry as a whole commits to purchasing OEM NGVs for their fleets. The U.S. gas industry, for example, purchases in excess of 5,000 pickup trucks per year which, if ordered as dedicated NGVs, could build an immediate market large enough to satisfy OEM's financial requirements to produce a production model NGV.[8] coupled with mandates such as those by the South Coast Air Quality Management District in California, entry of an OEM into the NGV market could easily become a reality.

The John Deere Company (Waterloo, IA) in conjunction with Clarke Power Products (Cincinnati, OH) is producing an 18,000 gross vehicle weight (GVW) chassis with a dedicated natural gas Ford engine, to be compatible with a variety of bodies including a 30 passenger bus, a "step-van" type delivery truck, and a beverage truck. This represents the first North American manufactured NGV with full warranty and a sophisticated distribution network in excess of 200 local dealerships. John Deere, in conjunction with A.G.A. is marketing the chassis with a utility truck body to gas utilities, and hopes to broaden sales to customers outside the utilities where dedicated NGVs could be used. Municipalities (using a bus body), airports, delivery fleets, beverage distributors and others could use the vehicle, providing that an adequate supply of compressed gas is made available to the customer.

The Gas Research Institute (GRI) has embarked upon a research and demonstration program including several diesel engine manufacturers, including Caterpillar and Stewart & Stevenson. The plans include development of a diesel retrofit kit to meet 1988 EPA diesel standards, and the development of a 100% dedicated, spark ignited diesel engine suitable for heavy truck and municipal bus applications. The program, which will put some $5 million or more into natural gas/diesel engine development, promises to produce some of the most significant NGV R&D to this point. The program is being coordinated with a $7 million Canadian Gas Association diesel and NGV development program, which should result in the marketing of natural gas/diesel engines (sometimes referred to a "giesel") that can meet the most stringent emission standards for particulates and NO_x. Coupled with some alternative fueled vehicle laws and regulations, this development could have a profound impact on NGV markets in North America and internationally.

Diesel Retrofit Activities

Though natural gas conversions of diesel engines is very much in the research and development stages, there are numbers of fleets now converting vehicles and testing their performance on dual fuel diesel/natural gas and dedicated natural gas in diesel engines.

- Atlanta Gas Light now has 1 diesel dump truck converted to dual fuel natural gas/diesel, and will be converting 14 more as part of their new natural gas/diesel program.
- Columbia Gas Co. has 19 diesel dump trucks operating, with good reports from operators about quiet, clean running vehicles.
- The Hamilton Street Railway Company in Hamilton, Ontario continues to run its 6 buses converted to on natural gas (a mix of dual fuel and dedicated engines) and plans to convert an additional 40 vehicles.
- Pierce Transit Company in Tacoma, Washington has two, two-stroke diesel engine buses running dual fuel on natural gas.
- Transit authorities are taking a more active interest in natural gas/diesel buses. A number of communities are looking at the possibilities of

converting transit buses to natural gas, including: the Los Angeles County Transportation Commission, which will convert 10 diesel buses; Tucson, which, after 1988 will be purchasing alternate fueled buses; and Phoenix, which has converted its first diesel transit bus.

Low Pressure Storage

Low pressure storage capability, using charcoal adsorbents in the cylinders at 500 pounds per square inch (psi) or less, would reduce the first cost of compression and possibly allow for redesigning the cylinder shape to fit more easily within a vehicle. Some testing has been done on vehicles and GRI, the Institute of Gas Technology (IGT), and a consortium of utilities working under the name of Future Fuels have been researching low pressure storage systems.

The first market entry of a low pressure storage cylinder and small compressor -- with a working pressure of approximately 275-300 psi -- is being introduced in 1988 by a company called G-Tech in Detroit, Michigan as a 'gas torch', a metal cutting tool with applications in scrap yards, automotive shops, and other businesses where metal cutting is essential. If successful, this market entry development could lead the way to automotive applications in the very near future.

Home Compressors

A small, high pressure home compressor is being introduced in 1988 in the Canadian market by a Swiss company called Sulzer. Working in conjunction with utilities, these units will be offered for lease to customers with NGVs for refueling at home, with the prospect for customers of rarely having to return to a traditional gasoline refueling station. If the advent of low pressure storage becomes a reality, the market for low pressure home compression could be a major boon to the gas industry, with a potential of doubling the load of a residence's load that installs such a system to refuel an automobile.

UTILITY INVOLVEMENT IN NGV MARKETING

Utility NGV development and marketing programs cover a broad spectrum of involvement, ranging from the benign, narrow-focus, "wait-and-see", do nothing approach to a much more aggressive, pro-active leadership role in the industry.

The Short-Sighted Approach to NGV Marketing

On the inactive end of the spectrum, utilities that view the NGV market in traditional financial terms see NGVs as not being competitive with low gasoline prices, and tend not to become involved in any NGV activities. Unconcerned about future market potential, they have not converted their own fleets to natural gas and, therefore, have little ability to market NGVs to customers. Even if customer opportunities do present themselves, the inactive utilities don't provide compressed gas and find it hard to compete against low gasoline prices. Furthermore, if utilities still run their own vehicles on gasoline they set a poor example for customers who often say to utilities, "If CNG is so good, why aren't the utility vehicles converted?"

The point can be made to these utilities about the virtue of running their fleet vehicles on the commodity they sell -- natural gas. Few gas utilities would heat their office buildings with oil-fired systems. Fewer -- if any -- gas utilities would use electric heat pumps as the heating/cooling system of choice. Yet these same companies are satisfied to run their vehicle fleets on a competitor's fuel -- gasoline. Even when natural gas as a vehicle fuel would be price competitive, if viewed in terms of the total capital investment for conversion equipment and compressors, many utilities would rather invest in alternative ventures that have traditionally higher returns on their investment. If all utilities viewed NGVs in these terms, NGV vendors in U.S. would be out of business, and there would be no future opportunity to respond to market forces and demand for alternate fuels when the economic environment improves -- particularly when gasoline prices again rise. If manufactured NGVs do appear off the assembly line, however, there could be a possible softening on the part of the non-active gas utility companies.

The Moderate Approach to NGV Marketing

Some utilities are supporting the NGV commercialization effort by maintaining their own company NGVs but not marketing to customers. Some companies, particularly in areas where gasoline prices are less than 30¢ per gallon differential with an equivalent gallon of natural gas, have been forced to retrench their marketing position yet continue to maintain an NGV fleet of their own. These companies still provide part of the backbone of utility support for NGVs, but must take a practical view of marketing realities -- as they see them -- without aggressively subsidizing customer conversions that otherwise could make NGVs attractive and economical. These companies are positioned, however, to jump back into the NGV market if market forces or environmental considerations make NGVs attractive.

The Aggressive Approach to NGV Marketing

The aggressive companies on the leading edge of the NGV marketing effort continue to convert their own fleets and those of their customers, while simultaneously working to develop an infrastructure of public refueling stations. Some examples are as follows:

NGV Conversions

- The Philadelphia Electric Co. will be converting an additional 70 company owned vehicles and are adding a compressor station. They now have 2 experimental diesel trucks converted to natural gas. One of their customers, the Funk Water Quality Company, A Division of Culligan Water Softener, Philadelphia, did its first 23 conversions in 1987 with a potential for about 100 more vans.
- Northern Indiana Public Service Company (NIPSCO) has 950 vehicles converted, representing the largest NGV fleet in the U.S.
- The Brooklyn Union Gas Company (BUG) now has 100 vehicles converted and will do about 100 more. Home compressors are being installed in two employees' homes. BUG will be providing two 100% dedicated natural gas buses to Command Bus Lines in Brooklyn, and is hoping to do a program involving 1,000 New York City vehicles.
- People's Gas Light & Coke Company, Chicago is converting an additional 65 vehicles.
- Atlanta Gas Light is converting 50 more utility vehicles, including conversion of 14 more diesel trucks, despite the fact that gasoline is priced only about $.20 higher than CNG.

NGV Refueling Stations

One of the principle strategies to making compressed natural gas available publicly is to have utility companies open their stations to NGV customers, thereby relieving a major financial burden for a customer who is about to embark on an NGV program. This refueling strategy is receiving more widespread support from the gas utilities that are marketing NGVs, and is likely to be the standard for the industry in making compressed gas available publicly while keeping the cost of the refueling station in the ratebase. Today there are 17 "semi-public" refueling stations in the U.S. All are utility or customer-owned with one serving CNG from a fuel dispenser as part of an existing service station.

- The first truly public refueling station opened (at an existing Amoco truck stop) by NIPSCO in Schererville, Indiana. Fuel is selling for $.65 an equivalent gallon.

- Wisconsin Gas Company opened its second public accessible refueling station at their south service center. Customers can drive in right off the street and use a computer card to obtain fuel. Wisconsin Gas also has developed a portable compressor/cascade system to bring to customers who want to have a test program for NGVs.

- Southwest Gas Corporation opened the world's largest NGV refueling station in Las Vegas, to serve company vehicles and customers who want to refuel. The facility can have a completely public refueling stations added outside the utility yard (should customer demand warrant it). The station includes four fast fill dispensers and numerous slowfill outlets, capable of refueling about 600 vehicles per day. Southwest Gas also has offered to build 18 public refueling stations in Phoenix and Tucson, subject to an agreement with the state to convert their vehicles. Southwest also will provide $500 per private vehicle converted if the state also provides $500 as a conversion incentive.

- Brooklyn Union and Con Edison are working on a deal to open 3 public refueling stations, including one at Kennedy International Airport. The plan is to use these stations to support the conversion of 1,000 New York City vehicles.

- Minnesota Gas Co. has opened public refueling stations in Minneapolis and Lincoln, Nebraska where they lease the compressor equipment to retail fuel operations.

- People's Gas Company in Pittsburgh opened 8 of their own compressor station facilities to NGV fleets in the area.

DEVELOPING A UTILITY NGV STRATEGY PLAN

Utilities have an opportunity to develop a strong strategy plan for NGVs in order to obtain a commitment to the NGV commercialization effort, using a multidisciplinary corporate approach to promote NGVs. The strategy plan must be action oriented, and is intended to:

- Develop a commitment to NGVs from within each functional group of the utility.
- Reach out to local decisionmakers involved with NGV efforts.
- Reach out to public interest groups (and especially the environmental community) including city managers and transit authorities.

The multidisciplinary utility approach to NGV marketing must involve all sections of the company, each of which has an important and specific role to play in the successful promotion of NGVs.

Utility Operations Section Actions

- Purchase OEM NGVs (John Deere, dedicated pickup and utility van).
- Convert vehicles (gasoline and diesel).
- Support opening of public and semi-public refueling stations throughout service territory.
- Support NGV R&D effort through improved recordkeeping of its own NGV fleet activities.

Marketing Actions

- Develop an NGV Strategic Marketing Plan.
- Build NGV demand for public refueling by promoting NGV use by customers (OEM purchases & conversions).
- Support and promote transit bus demonstration program.

Government Relations Actions

- Work with A.G.A. Federal lobbying efforts to executive and congressional branches of government (including development of NGV emission standards that includes reactive/non-reactive hydrocarbon standard and 5 year retest period of CNG cylinders).
- Develop lobbying effort for favorable state and local alternative fuels legislation & programs including: support for demonstrations; provide tax break for CNG as a fuel; use OEM and converted NGVs; removed sale-for-resale prohibitions (where applicable).

Communications & Public Relations Activities

- Promote NGV awareness events with marketing and government relations groups.
- Issues news releases about NGV industry events.
- Develop case studies of NGV success stories, and promote the concept and technology with national advertising program.

Finance and Legal Actions

- Develop NGV rate to sell gas.
- Work to ratebase the cost of compressor stations.
- Develop a more favorable approach with the insurance industry to insure NGV refueling facilities.

Any utility that takes a coordinated, aggressive and far-sighted approach to NGV marketing as described, can be assured of a future market and can make a major contribution to helping to solve air pollution problems within their communities.

NOTES.

1. Monthly Energy Review, Energy Information Agency, U.S. Department of Energy, Washington, D.C., volumes through 1986-1987.

2. Gas Industry Five-Year Strategic Marketing Plan, American Gas Association, Arlington, VA, Successive annual volumes from 1984-1987.

3. The Clean Air Act 42 U.S.C. 7401 *et seq*.

4. *Assessment of Costs and Benefits of Flexible and Alternate Fuel Use in the U.S. Transportation Sector, Progress Report One: Context And Analytic Framework*, U.S. Department of Energy, DOE/PE 0080 January 1988.

5. *Ibid.*

6. *The NGV Reporter: Product & Service Directory*, American Gas Association, Arlington, VA, January 1988.

7. Private conversations with Ford & GM.

8. *1986 A.G.A. Fleet Survey*, American Gas Association, Arlington, VA, December 1986.

SECTION

XXIII

STATUS REPORT ON NGV TECHNOLOGY

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Bill Hennessey
B. C. Hydro

9/5/666

ABOUT THE SPEAKER:

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For the past five years, he has been working solely in the Natural Gas Vehicle industry with B. C. Hydro and in both technical and marketing capacities.

Bill has been involved in CGA activities, and has worked with suppliers, manufacturers, technical schools and universities. From these activities he has obtained broad technical knowledge about the use of natural gas in combustion engines.

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STATUS REPORT ON NGV TECHNOLOGY

By: W. G. Hennessey, P. Eng.
B. C. Hydro

Natural gas has been used worldwide for decades in hundreds of thousands of vehicles. The technology required for effective reliable low cost vehicle conversion components is really quite basic. The equipment needs only to ratio two vapours - air and natural gas, in a 10 to 1 volumetric ratio. This is far easier than ratioing a liquid (gasoline) with a vapour, air, primarily because of the temperature and pressure dependance of the vapourization rate of gasoline.

For that reason natural gas vehicle conversion equipment changed very little over the years and, compared to the average carbureted gasoline system, did a better job over the life of the vehicle.

Then along came gasoline fuel injection and computer control. How would the old natural gas technology meld with the new gasoline technology, and what technological development is needed to use natural gas effectively and efficiently in large vehicles such as trucks and buses, and possibly in locomotives.

Those questions were not being answered in Italy, New Zealand or the USA, the areas of greatest NGV activity, so the Canadian Gas Utilities, through the Canadian Gas Association, decided to develop a Research and Development program and to obtain R&D funding. As a result, projects valued at $7.2 M have been approved or committed for 54 projects and a further $1.8 M are budgeted for 6 additional projects.

All projects were intended to support three major marketing objectives: use of NGV in diesel-engined buses and trucks; improved conversion kits for spark ignition engines, particularly in new vehicles; and faster, more efficient refuelling. In addition, the program supported safety research and technology transfer. While each project is autonomous, individually budgeted, with specific objectives, most are interrelated.

This R&D program is still in its infancy, so few projects have been completed, or final reports written. However, there have been significant findings to date and commercialization of certain products shall soon occur.

This is not to suggest that NGV research has not been done prior to the CGA program. On the contrary, B. C. Research maintains a library of NGV reports of considerable magnitude. However this research has not been targeted for or directed towards commercialization of NGV products.

The CGA research will build upon this valuable resource of information, and result in technology, whether on paper or in hardware will be in demand worldwide.

DIESEL ENGINE PROGRAM

In the diesel engine program, four major questions had to be answered:

1. Can fuel system/engine control systems achieve acceptable driveability, emissions, and economics for NGV? If so, which technology approach matches which application?

   Four projects are underway to answer that question.

   ### Diesel Baseline Studies

   The project, contracted to B. C. Research and UBC, is to explore the fundamental parameters that affect performance of a direct injected dual-fuelled diesel.

   The first phase involves use of a single cylinder Ricardo engine and the second phase involves multicylinder engines.

   ### Dual Fuel Diesel/NG Engine Development

   This project being conducted by B. C. Research and Natural Energy Research Ltd., with cooperation of Caterpillar, involves developing a strategy and hardware to operate a computer controlled 450 hp Caterpillar 3406 B engine on natural gas and diesel.

   This involves developing a computer which will talk to and work with the Caterpillar computer.

   ### Heavy Duty Otto Cycle Engine Optimization

   No work has been done on this project yet, but the Ontario Government has funded a similar project to develop a spark ignition version of the Cummins L-10 engine.

   ### Large Dual Fuel Diesel/NG Engine Development

   Transport Canada, Gaz Metropolitan, Ontario Ministry of Energy and Bombardier are finalizing funding to convert a Bombardier 12-cylinder 251 locomotive engine to operate at up to 95% natural gas, 5% diesel fuel ratio.

2. What market segments do we target and what are their characteristics?

   A Diesel Technical Assessment Study is being carried out to determine the appropriate engines and markets for which R&D should be directed.

   Initial results indicate that the school bus, urban bus and commercial trucking are the target markets.

3. Can the low flame speed characteristics of natural gas be compensated for in NGV engine design?

   Project 1.3 conducted by UBC and the University of Victoria is to study the effects of enhanced turbulence, combustion chamber design and high energy ignition processes in spark ignition engines.

4. What measures are required to NGV engines to obtain levels of durability and reliability comparable to those of diesel engines?

   Accelerated wear testing of NG-powered diesel engine blocks will be carried out on a Cummins L-10, 200 hp engine by the Ontario Research Foundation in cooperation with Cummins Engine Co. and on a Caterpillar 3406 B by B. C. Research in cooperation with Caterpillar.

SUMMARY OF DIESEL PROGRAM

Results of testing to date appear to indicate that both dual fuel and Otto Cycle approaches are viable, that OEM interest is on the rise and that extensive field demonstrations will become viable from 1 to 3 years. These projects are answering some of the unknowns in this field.

GASOLINE ENGINE PROGRAM

In the gasoline engine program three areas were to be investigated:

1. Improve existing NG carburetion equipment.

   Three projects are underway to address that area:

   - Upgrade existing underhood technology for improved cold starting, idling, OEM air intake systems compatibility and ease of calibration.

   - Develop efficient methods for mounting an oxygen sensor in the exhaust for better calibration.

   - Develop a digital, programmable ignition timer for use on all engines, including those with distributorless ignition.

2. Develop micro-computer based natural gas fuelling systems.

   Air Sensors Inc. of Seattle has been contracted to develop such a system, based on a mass flow measurement. One prototype is now operational.

3. Determine the measures required to obtain gasoline engine durability and reliability.

   The Ontario Research Foundation has conducted some work in this area.

FUELLING SYSTEMS

Four areas of fuelling system R&D are being conducted.

1. The large refuelling station.

   - Union Gas and B. C. Hydro Gas are evaluating and testing the most efficient method for refuelling a fleet of single fuel natural gas urban transit buses.

     The prototype system will probably be used by the bus system in Hamilton, Ontario, where dedicated natural buses are now in scheduled service.

   - B. C. Hydro and B. C. Research are flow testing refuelling system components and developing a computer program to optimize the design of public station NG refuelling systems.

   - A data base of existing station operating data is being collected.

2. The home and commercial refuelling appliances.

   - Installation, operation and maintenance data is now being collected on a home refuelling appliance installed at locations across Canada.

   - A farm demonstration program is being conducted in Saskatchewan using a commercial compressor and a number of converted farm vehicles.

   - Various filling probe designs are being studied for different applications.

   - B. C. Hydro Research and a number of subcontractors are conducting a study to optimize the design of vehicle storage cylinders.

     Four major manufacturers of vehicle cylinders are participating, covering all technologies including the all metal, the hoop-wrapped metal, and the all fibre cylinders.

A method to recertify cylinders without removing them from the vehicle will also be studied.

- B. C. Hydro Research also has a contract to design tests which can lead to reduced cost and weight of the existing generation of cylinders.

That completes my summary of the Research and Development efforts presently underway.

As Canadian technology progresses as a result of this R&D, there will be further commercialization opportunities inside and outside Canada, and possible, exports of Canadian NGV technology.

SECTION

XXIV

SMALL-SCALE FUELING SYSTEMS

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by Gordon Bryce
B. C. Hydro

9/5/666

ABOUT THE SPEAKER:

Gordon D. Bryce was born in Vancouver and obtained a B.Sc. in Mechanical Engineering in 1970 at the University of British Columbia. He joined B. C. Hydro in May 1974.

His initial work was in pipeline design specializing in bridge crossing and underwater surveys.

Gordon joined Hydro's Natural Gas for Vehicles Group in October 1984, where he is involved with retail service station design, marine applications, and home compressors.

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SMALL SCALE FUELLING SYSTEMS

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1.0 INTRODUCTION

At B. C. Hydro, we have put most of our effort on small scale fuelling systems into the residential application. For this reason I will devote most of my discussion to this aspect. However, there are obvious commercial applications for small scale fuelling systems and I will touch on these.

This paper will deal primarily with the design of the complete fuelling system rather than simply the compressor. The system must be physically and economically compatible with the user, the surroundings and the vehicle.

Also included in this paper is a brief description of the history of small scale fuelling systems at B. C. Hydro, the present situation, the expected impact on vehicle conversions and possible future developments.

2.0 DEFINITION OF A SMALL SCALE FUELLING SYSTEM

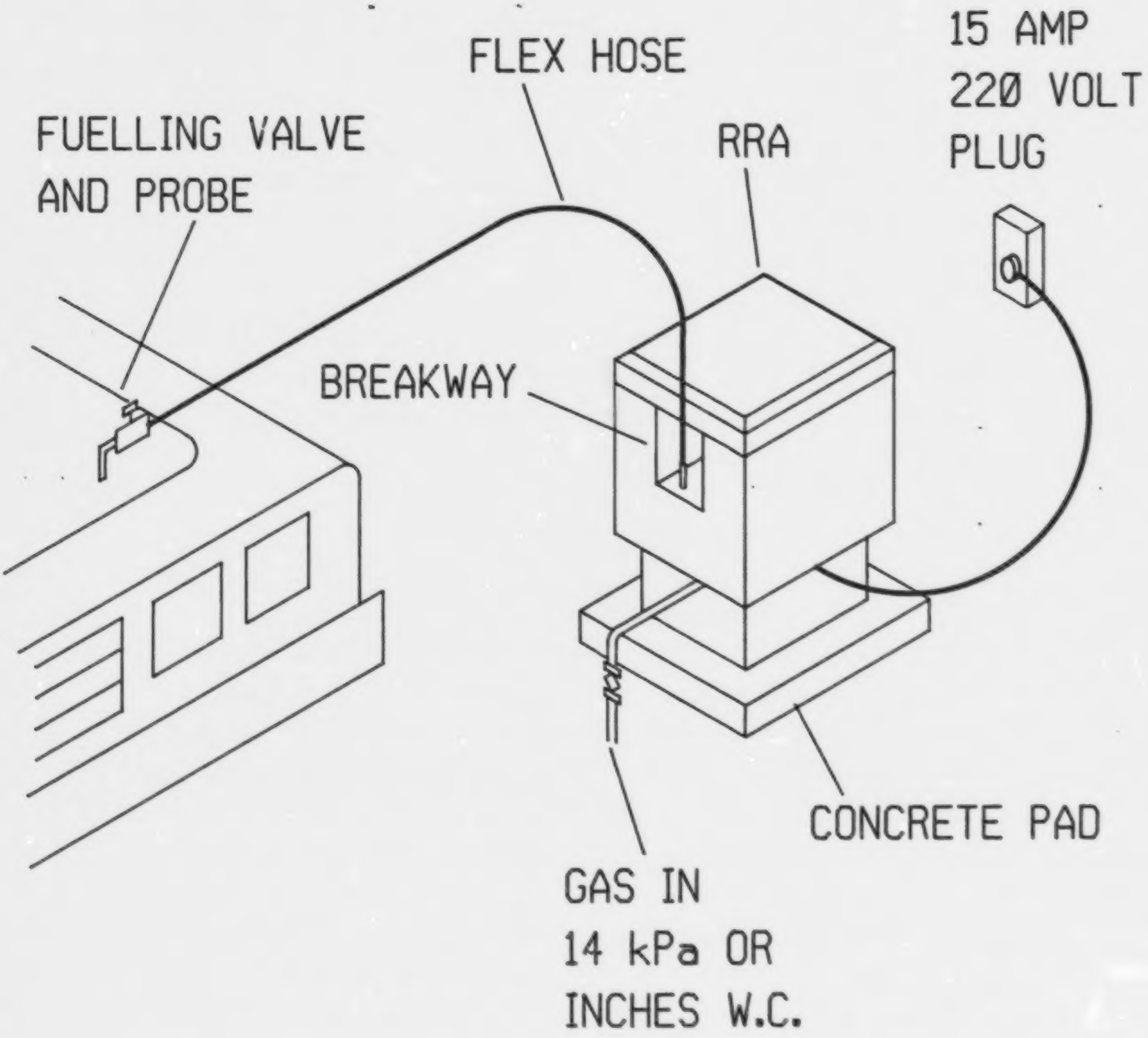
The regulations governing natural gas installations, CGA B149.1-M86, specifically do not apply to "home compressors". The term home compressor has gone out of fashion and been replaced, in most people's minds with "Residential Refuelling Appliance" or RRA. The CGA Laboratory Test Requirement for RRA's stipulates a capacity of 10 scfm or less. This provides a starting point to define a Small Scale Fuelling System.

In a residential application, there is no need for that much capacity. Indeed the constraints imposed by the typical residential electrical system restrict the appliance's capacity to approximately 2.5 scfm.

The CGA Laboratory Test Requirements do not address on-site storage. It is likely that storage and storage control systems will not be installed in most small scale fuelling systems in

FIGURE 1

TYPICAL SMALL SCALE FUELLING SYSTEM


FLEX HOSE
15 AMP
220 VOLT
PLUG
FUELLING VALVE
AND PROBE
RRA
BREAKWAY
CONCRETE PAD
GAS IN
14 kPa OR
INCHES W.C.

ERRA1 NGVRRA.DGN

order to keep-it-simple. For residential applications the municipal authorities may rule out on-site storage. One major advantage of natural gas as a vehicle fuel is that home refuelling can be carried out without storing an appreciable amount of fuel at the home.

A Residential Refuelling Appliance is a low capacity, low powered compressor taking natural gas from the residence's housepiping system and dispensing it directly into a vehicle (Figure 1). In addition, the RRA must be quiet, reliable, self-protecting, convenient and inexpensive. Surely that is not too much to ask.

Small scale commercial fuelling systems will likely employ a RRA and be very similar to residential installations.

3.0 RESIDENTIAL APPLICATIONS

The RRA must be compatible physically and economically with the user, the environment and the user's vehicle.

3.1 Physical Compatibility

3.1.1 Compatibility with the User

The RRA must be as simple, safe, reliable and convenient to use as any other gas appliance. In effect it must become "transparent" to the user. This means that the user must not need to inspect, adjust or maintain the appliance. The appliance must have enough built-in controls to safeguard itself and its surroundings.

The appliance must be convenient to use. As I will discuss later the appliance is expected to be cost neutral for the user. Therefore, the incentive to have and to use the appliance will need to be savings in other areas or in the convenience of fuelling, hopefully both. I have been using a RRA at my office for a few months and I have the connect and disconnect operations down to 30 to 45 seconds each. This is a significant improvement over the 7 to 10 minutes I would likely spend refuelling the vehicle at a retail station.

3.1.2 Compatibility with the Surroundings

Ideally a RRA should be out of sight and out of mind. However, the need to place the RRA near the refuelling vehicle makes this difficult if not impossible. The RRA installation should be as unobtrusive as possible, blending into the residence's architecture if possible or tucked into the landscaping. Also the RRA's housing should disguise its purpose. An industrial compressor setting on the front lawn would not be attractive to most people.

The appliance will likely be running for part of the night. It must be quiet enough to not disturb the user or the neighbours or infringe on municipal noise bylaws.

The appliance must not release gas during normal operation. The smell of gas can be a nuisance to the user, the neighbours and the gas utility.

The appliance must not disturb the other gas appliances. Also the appliance must operate on normal household electrical service, i.e. 110 or 220 V single phase.

3.1.3 Compatibility with the Vehicle

The appliance must have enough capacity to support the normal daily activity of the user's vehicle. There is little or no benefit in having extra capacity particularly as extra capacity would likely increase the cost of the appliance and installation.

The appliance must be capable of putting out the correct pressure. Too much pressure will obviously endanger the vehicle fuel storage cylinders. Too little pressure would result in a reduction in the vehicle's range. Range is a problem with natural gas powered vehicles and may become critical when a RRA user tailors the vehicle storage to the normal daily requirement.

Ideally the appliance will not allcw any oil carry over in the gas stream. Any oil that gets into a vehicle fuel system can cause erratic performance. Removing oil from a vehicle's fuel system can be expensive.

The connection between the appliance and the vehicle must be compatible with the existing fuelling probe or a second special fitting must be installed. Refuelling at a retail station must remain an option for the user. The connection must include a flexible hose and a breakaway. The breakaway will disconnect the fuelling line if the vehicle moves away from the appliance while the fuelling line is still connected. The fuelling hose must contain a valve of some sort; however, the three-way valve used at the retail stations is not required where the fuelling line bleeds down into a recovery vessel.

A typical residential installation is shown on Figure 1.

3.2 Economic Compatibility

At B. C. Hydro, we expect that a RRA in a typical residential application will be cost neutral for the user. That is the cost of fuel from a RRA will be nearly the same as from a retail outlet.

If there is no fuel cost savings from a RRA where is the incentive to have and use one? How can B. C. Hydro determine the fuel price?

3.2.1 Economic Incentive to Have and Use a RRA

The RRA will be more convenient than public refuelling as mentioned before.

The economic incentive to have and use a RRA is a reduction in the vehicle conversion costs. Because the user can depend upon refuelling conveniently every night the vehicle's fuel storage can be tailored to the user's daily needs. This will in many cases allow a single storage cylinder to be used in place of the two cylinders commonly used now. This could result in a $500 to $800 savings. To put this in perspective this would be a 20% reduction for most conversions.

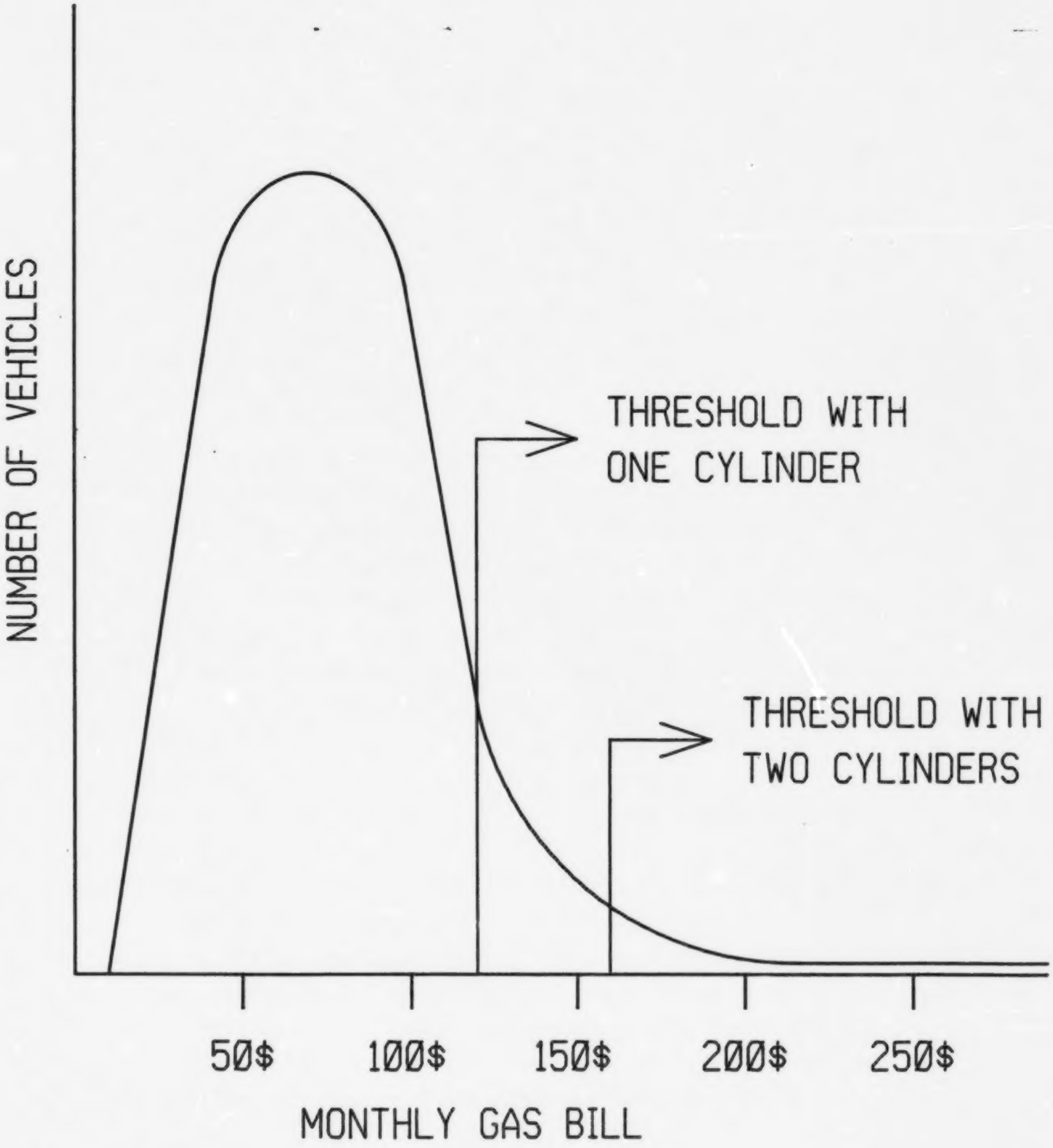
The NGV market has matured enough that the decisions are usually economically driven. The people converting now are not pioneering gadget freaks and few people are scared off by the novelty of the concept. The 20% reduction in conversion costs will attract a new group of NGV users. The threshold gasoline consumption that makes NGV an attractive choice could drop from $160 per month to $120 per month. The new group of potential NGV users include all those drivers with monthly gasoline bills between $120 and $160. The size of this

FIGURE 2

(HYPOTHETICAL)

DISTRIBUTION OF VEHICLE FUEL CONSUMPTION

GASOLINE COST PER MONTH VS NUMBER OF VEHICLES



ERRA2 NGVRRA.DGN

group depends upon the density distribution of fuel consumption vs number of vehicles. Figure 2 shows a hypothetical distribution curve. This figure illustrates two points: the first is the new group is larger that the existing group of converted vehicles; the second is this group still does not include the average driver.

3.2.2 Economics of RRA Ownership

A typical RRA installation is expected to cost about as much as a vehicle conversion. However, the economic benefit of a RRA is expected to be less than that of a vehicle conversion. Only very high fuel consumption drivers could rationalize the purchase of a RRA, and these drivers would likely already be using NGV.

The economics of RRA ownership are subtly better for a utility. For this reason the utilities are likely to own the RRA's and rent or lease them to the users. The utility can then structure the tariff so that the fuel cost is nearly the same as that at the retail station.

The municipal officials have expressed a strong desire to see the utilities maintain ownership and control over the appliances.

4.0 COMMERCIAL INSTALLATIONS

As noted before, a RRA may have a capacity of 10 scfm, enough to support a fleet of 10 vehicles. The appliance and the fleet would have to match very well. The ideal fleet vehicle would operate 8 to 10 hours per day, consume an average of 22 Le per day and return to a base each day. Under these conditions a 10 scfm appliance could support a 10-vehicle fleet. Higher average fuel consumption would reduce the number of vehicles. If less stationary time is available a fast-fill system with higher costs and greater complexity would be required. I may be difficult to convince provincial or municipal authorities that a fuelling system that includes storage is an appliance.

Other than small fleets a RRA could be used to support vehicles such as forklifts, lumber carriers and zambonis (ice resurfacing machines) which cannot be refuelled at a retail station.

5.0 HISTORICAL DEVELOPMENT

The RRA would have been the ideal device to introduce the concept of natural gas as a fuel for private vehicles. The private vehicle market was difficult to establish because no one would convert their vehicle unless the fuel was readily available. Also, no one would install fuelling facilities unless there was a sizeable vehicle population. The RRA could have solved the chicken-and-egg problem by providing a dedicated fuelling system for each converted vehicle.

B. C. Hydro looked into possible RRA's in 1982 and came up with one reasonable candidate. We purchased a Rix Hom-Gas compressor and installed it in our Burnaby facility. However, by the time the compressor was installed the chicken-and-egg problem was well on its way to being solved. The effort required to get the early retail network installed interferred with our efforts to pursue the RRA. As a result of this interference and our position at the bottom of the learning curve, the Rix languished in our back yard until the spring of 1987 when Rix Canada renewed their interest and began to modify and upgrade our compressor.

In the spring of 1986 there was a completely unprecedented and until now unparalleled occurrence. A prototype home compressor arrived completely unannounced. I had about five minutes warning to clear a space on my layout table for it to rest on. This was a modified Scuba air compressor from Aero Technica Coltri of Italy. We tested the compressor for the next year or so and learned a great deal.

A few months later another prototype small natural gas compressor arrived from Italy. A slow speed hydraulic compressor from SAFE. At 5 to 8 scfm the compressor's capacity was both too large and too small. Too large for the residential application and too small for the typical commercial application as we saw it then. We test ran the unit for six months putting nearly a thousand hours on it before it had to be removed and returned to Italy. SAFE has taken the lessons learned during that testing and has produced a little brother to the original compressor. This little brother has just recently arrived and should be available for testing by a utility soon. Hopefully, that utility will be B. C. Hydro.

B. C. Hydro's role in developing a RRA has been to encourage and advise the various manufacturers and to do some preliminary testing on prototype compressors and appliances. To this end we have talked to Rix, Coltri, SAFE as previously mentioned, also Klassy Compressors from New Zealand, Dynamax Energy Corporation,a very bright and energetic local company, as well as Sulzer Brother of Switzerland.

We first talked to Sulzer just over two years ago and have met with them frequently. They have quietly developed their technology and tested their compressor and appliance package leading to an unveiling in August 1987. This package has now become the Sulthane RRA which is being installed across Canada in coordinated field trial programs by most of the major gas utilities. Similar programs are also underway in Australia and New Zealand.

We feel very positive about the future of RRA's with the various manufacturers progressing through the logical development stages.

6.0 PRESENT CONDITION OF RRA DEVELOPMENT

At the present time, B. C. Hydro is participating in a field trial program which is a cooperative effort by B. C. Hydro, Sulzer, the Provincial Gas Safety Branch, the Provincial Fire Marshall's Office and the Municipality of Surrey. In this program, we will install up to 8 Sulthane appliances at private residences. The purpose of this program is to get some real world operating information. This information includes installation and operating costs, operator reaction, impact on the operator's neighbours, sound levels and operating problems.

This program is part of the Canada-wide effort mentioned before. However, in order to help manufacturers we will be performing a similar field trail program for any appliance that we consider appropriate. We will perform tests on the appliances at our Burnaby facility to determine their appropriateness.

As a result of this first field test program we hope to be able to assist the CGA or the Provincial Gas Safety Branch in writing manufacturing and installation standards for RRA's. We also want to have the municipal authorities become familiar and comfortable with the appliances so that their concerns can be addressed in the installation code. This will avoid having a hodgepodge of municipal bylaws interferring with the orderly development of the RRA market. Once the standards have been written, B. C. Hydro should be able to eliminate the field trial program.

A marketing study is underway by the CGA to determine what marketing strategies are likely to be successful in introducing the RRA.

Once the field trial program and the marketing study are complete the next step will be to test market the appliances. This test market will likely occur in the same area as the field trial program; in this case, the Municipality of Surrey. The test market will refine the proposed marketing strategies and establish a likely market penetration.

7.0 FUTURE DEVELOPMENT

The RRA as it stands now has the potential to dramatically increase the number of natural gas powered vehicles. The RRA by itself will not make natural gas the logical economic choice for the average private motorist.

It will require a break through of some sort to make natural gas economically attractive to the average motorist. Such a break through would be another oil crisis, an original equipment manufacturer supplied dual fuel vehicle or perhaps adsorbtive storage.

Adsorbtive storage would be particularly beneficial for the RRA market. If a vehicle could have adequate range at a maximum pressure of 500 psig the cost and power consumption of a RRA could be greatly reduced.

SECTION

XXV

CASE STUDY - FUELING STATION FOR DIESEL VEHICLES

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B. C.
13 - 14 April, 1988

by John Adams
John F. Adams Ltd.

9/5/666

ABOUT THE SPEAKER:

Mr. Adams was born and educated in the Lower Mainland of British Columbia.

He has been involved in the Marketing of Petroleum Products for Industrial and Vehicular applications since 1949. He established his own company in 1959, which was the agent for a national oil company.

In 1970 he established a new independent distributorship for vehicle fuels and lubricants. The first three mechanical keylock fuel installations in the area, were part of this operation. In 1980 the company converted, for its commercial fuel operation, from mechanical keylocks to electronic card-operated fuel dispensing.

Mr. Adams became interested in alternative fuels in the 70's and in 1985 started to develop the first Commercial Natural Gas Station specifically designed for all sizes of commercial trucks (Class 8). The new fuelling facility was put into operation in the spring of 1987. The company was assisted in this venture by the Department of Energy, Mines and Resources, Ottawa, and B. C. Hydro - Gas Division. This facility is a component of a Demonstration Project which also involves the conversion of diesel engines to a dual fuel mode.

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9/5/652

I HAD BEEN INTERESTED IN ALTERNATIVE FUELS FOR MANY YEARS AND IN THE 1970'S, IN LINE WITH OUR COMPANY'S POLICY OF TRYING TO BE UP-TO-DATE IN FUEL TECHNOLOGY I STARTED RESEARCHING ALTERNATIVE FUELS FOR OUR COMMERCIAL VEHICLE FUEL STATION. WE PUT IN PROPANE IN 1981 AND IN 1985 COMMENCED AN INVESTIGATION OF NATURAL GAS FOR VEHICLES.

IT WAS RECOGNIZED THAT NATURAL GAS FOR VEHICLES WAS A FUEL WITH SOME APPLICATION NOW BUT WITHOUT A DOUBT, IN MY OPINION, THE ONLY ONE OF THE NOW KNOWN ALTERNATIVE FUELS THAT SHOULD BE SERIOUSLY CONSIDERED IN THE CANADIAN FUEL SCENE. IT IS LOW COST, WIDELY AVAILABLE AND SAFE.

WHILE ALTERNATE FUEL SUBSTITUTION IS REALLY CRITICAL TO CANADA WITH OUR KNOWN SMALL RESOURCES OF CONVENTIONAL FUEL IT REQUIRES A SERIOUS SHOVE BY GOVERNMENTS AND UTILITIES TO GET NATURAL GAS USED AS A SUBSTITUTE FOR DIESEL FUEL.

WHEN YOU CONSIDER THAT IN ONE YEAR A CAR USES 1 1/2 TIMES THE NATURAL GAS THAT A HOME USES AND THAT A CLASS 8 HIGHWAY TRUCK USES 12 TIMES WHAT A HOME USES IT IS VERY OBVIOUS WHERE THE ALTERNATIVE FUEL DEVELOPMENT EMPHASIS SHOULD BE. I'LL STOP EVANGELIZING NOW AND TELL YOU ABOUT OUR CARD- ACCESSED NATURAL

GAS PROJECT. OUR PROJECT'S PURPOSE IS TO DEMONSTRATE THE TECHNICAL AND ECONOMIC FEASIBILITY OF NATURAL GAS FOR DIESEL ROAD APPLICATIONS.

IT WILL PROVIDE DESIGN AND OPERATIONAL EXPERIENCE, FIELD OPERATION DATA AND FEEDBACK FOR R&D PURPOSES AND A "NEEDED" WORKING EXAMPLE TO SPUR FURTHER DEVELOPMENT OF THE INDUSTRY.

THE PROJECT INCLUDES THE FOLLOWING:

(1) NGV/DIESEL FUELLING STATION DESIGN AND INSTALLATION
(2) CONVERSION OF DIESEL ROAD VEHICLES
(3) PROGRAM MONITORING AND DATA COLLECTION
(4) TECHNOLOGY TRANSFER THROUGH PREPARATION OF REPORTS AND TECHNICAL PAPERS OVER THE COURSE OF THE PROJECT

I'LL FIRST DEAL WITH THE NGV/DIESEL FUELLING STATION DESIGN AND INSTALLATION. B.C. HYDRO - GAS DIVISION PROVIDED SUBSTANTIAL ASSISTANCE IN THE DESIGN OF THE FACILITY. THE ENTHUSIASTIC SUPPORT WE RECIEVED FROM THEIR MARKETING PEOPLE AND THEIR ENGINEERING DEPARTMENT WERE A MOST VITAL INGREDIENT TO THE SUCCESSFUL COMPLETION OF THE STATION. IT WAS NECESSARY FOR MY COMPANY TO AQUIRE ADDITIONAL LAND TO EXPAND OUR EXISTING FACILITY TO ACCOMODATE THE PROJECT. RECOGNIZING THAT, OVER A PERIOD OF FIVE YEARS, IT WAS REASONABLE TO EXPECT THE NEED FOR AN INCREASE IN THE SIZE OF THE FACILITY, WE PUT IN THE PILING AND SLAB TO ENABLE US TO INCREASE

THE AMOUNT OF COMPRESSION BY ADDING MORE COMPRESSORS OR A LARGER COMPRESSOR. IT, OF COURSE, FOLLOWS THAT WE WOULD NEED INCREASED STORAGE SO ADDITIONAL PILING, SLAB STRENGTH AND SHELTER WAS INSTALLED TO ACCOMODATE SIX MORE A S M E CYLINDERS. IT WAS RECOGNIZED THAT LARGER PIPING WOULD BE NECESSARY IN THE FUTURE, SO THAT PIPING WAS INSTALLED FOR AN ADDITIONAL DISPENSER WHICH WOULD BE DESIGNED TO FUEL FASTER THE EXPECTED LARGER AMOUNT OF STORAGE ON TRUCKS. ALL NECESSARY WIRING WAS ALSO RUN UNDERGROUND FROM THE MOTOR CONTROL CENTRE TO THE PUMP ISLAND AND TO THE COMPRESSOR SLAB TO AVOID DISTURBING THE PRESENT INSTALLATION AT THE TIME OF INCREASED CAPACITY CONSTRUCTION. THE COMPRESSOR SELECTED WAS A PRESCON MODEL 3D360/6/100R DUKE 223 C.F.M. MANUFACTURED IN NEW ZEALAND BY PRESCON AND SUPPLIED AND INSTALLED BY INTERNATIONAL PRESSURE CONTROL SYSTEMS INC. OF BURNABY. THE STATION LAYOUT WAS DESIGNED BY OURSELVES BASED UPON MANY YEARS OF EXPERIENCE AND IS ABOUT AS GOOD AND FUNCTIONAL AS POSSIBLE. THE CARD CONTROL SYSTEM WAS MADE BY COMPUTROL, A VANCOUVER COMPANY. ALL THE EQUIPMENT WORKS WELL WITH AN ABSOLUTE MINIMUM OF TROUBLE.

***** SLIDES

DEALING WITH THE IMPLEMENTATION OF THE SECOND COMPONENT OF THE PROJECT, NAMELY THE CONVERSION OF DIESEL ENGINES TO USE EITHER NATURAL GAS WITH DIESEL (DUAL FUEL) OR NATURAL GAS ALONE (A DEDICATED ENGINE); WE ARE NOW INTERVIEWING PROSPECTIVE VEHICLE OPERATORS TO ESTABLISH A "FIT" WITH OUR ENERDEMO PROGRAM. THIS PROGRAM INVOLVES THE SUBSIDIZED CONVERSION OF THE CUSTOMERS ENGINE. THE NUMBER OF CONVERSIONS HERE AND ABROAD WOULD PROBABLY TOTAL IN THE LOW THOUSANDS. TECHNOLOGY DEVELOPMENT HAS BEEN HAPPENING IN SOMEWHAT ISOLATED POCKETS IN THE WORLD AND WITH MANY DIFFERENT ENGINES, MANY OF WHICH ARE NOT COMMON IN THIS AREA. A GREAT DEAL OF THE WORK HAS BEEN DONE IN NEW ZEALAND BY BILL SHIELDS OF TRANSPORT FUEL SYSTEMS (NZ) LTD. FOR INSTANCE, AT PRESENT HE IS CONVERTING A DEDICATED CUMMINS 855 ENGINE FOR DEMONSTRATION IN FORT ST. JOHN FOR DWIGHT CLARKE AND ICG. LOCALLY. B.C. RESEARCH HAS DONE DEVELOPMENT WORK ON A CATERPILLAR 3208 ELECTRONIC SYSTEM. TONY FILETTI OF MOGAS FUEL SYSTEMS INC., JOE FOSTER OF PRO STAFF FUELS AND ANKER GRAM OF CRYOGAS ENGINEERING LTD. HAVE ALL DONE A FEW SUCCESSFUL CONVERSIONS. AS PROJECT MANAGERS FOR THE E M & R ENERDEMO PROGRAM, OUR COMPANY MUST BE ABSOLUTELY SURE THAT ONLY APPROPRIATE ENGINES ARE CONVERTED WITH SURE TECHNOLOGY.

FROM WHAT I AM ABLE TO FIND OUT ABOUT DIESEL ENGINE CONVERSION TECHNOLOGY MUCH MORE MONEY MUST BE SPENT IN R & D.

MOST PEOPLE INVOLVED IN THE CONVERSION TECHNOLOGY URGENTLY REQUIRE MUCH MORE FINANCIAL SUPPORT. THERE IS ALSO AN URGENT REQUIREMENT FOR THE SHARING OF CONVERSION TECHNOLOGY INFORMATION. WHEN THERE ARE ONLY A HANDFUL OF PEOPLE INTERNATIONALLY IN THIS DEVELOPMENT WORK IT IS INCUMBENT UPON THE PRINCIPAL BENEFICIARIES, NAMELY UTILITIES AND GOVERNMENTS TO BE SUBSTANTIALLY MORE SUPPORTIVE. I AM CONVINCED THAT THE AGRESSIVE DEVELOPMENT OF NATURAL GAS AS A SUBSTITUTE FOR DIESEL FUEL IS EMINENTLY SENSIBLE IN TERMS OF R O I, IN REDUCTION OF DEPENDENCE ON CONVENTIONAL FUELS AND IN REDUCTION OF POLLUTION IN OUR AIR - PARTICULARLY IN CITIES.

ANOTHER THRUST WHICH WILL BE REQUIRED IS THE DEVELOPMENT OF A FUELLING NETWORK THROUGHOUT CANADA. THIS COULD PROBABLY BE ACCOMPLISHED RELATIVELY EASILY BY THE UTILIZATION OF COMMERCIAL CARDLOCK NETWORK PROPERTIES WHICH NOW EXIST ALL ACROSS THE PROVINCES. AS AN INTERIM STEP, WHEN OIL COMPANIES AND GAS UTILITIES ARE PLANNING EXSPANSION OF NATURAL GAS FUELLING FACILITIES IN CONVENTIONAL SERVICE STATIONS, IT WOULD BE SENSIBLE TO PLACE THE DISPENSERS, IF SPACE PERMITS, IN A LOCATION THAT

COULD BE ACCESSED BY TRUCKS. HAD THIS BEEN DONE IT WOULD HAVE BEEN MUCH EASIER TO MATCH TRUCKS TO REFUELLING AT INTERIOR B.C. DESTINATIONS FOR PROSPECTIVE ENERDEMO PROGRAM PARTICIPANTS.

IN ALL MY READING OF MATERIAL FROM ALL OVER THE WORLD, FROM THE 1986 CONFERENCE IN VANCOUVER AS PART OF EXPO, FROM THE AGA SEMINAR IN INDIANAPOLIS LAST SEPTEMBER AND TRAVEL TO NEW ZEALAND I AM CONVINCED THAT WE ARE WITHIN A VERY FEW YEARS OF URGENTLY NEEDING PROVEN DIESEL ENGINE CONVERSION AND FUELLING TECHNOLOGY. WHY? BECAUSE DIESEL CONSUMPTION IS INCREASING EVERY YEAR AND WORLD CRUDE PRICES WILL SURELY INCREASE DRASTICALLY WITHIN TWO YEARS MAXIMUM. REMEMBER, OPEC WAS FORMED IN 1960 AND THE PRICE PER BARREL THEY SET WAS $1.80. IN 1971 THE PRICE WAS $2.18, IN 1973 - $5.12, IN 1974 - $10.95, IN 1979 - $14.59, IN 1981 - $35.00, IN 1983 - $33.00. THIS RECENT HISTORY SUGGESTS THAT WE ARE TOO COMPLACENT IN OUR DEVELOPMENT OF ALTERNATIVE FUELS.

IN SPITE OF THE RELATIVELY SMALL NUMBER OF PEOPLE INVOLVED I HAVE OBSERVED A STEADY PROGRESS IN CONVERSIONS AND CONVERSION TECHNOLOGIES OVER THE LAST THREE YEARS. IN CANADA THE TORONTO TRANSIT COMMISSION IS NOW PROCEEDING WITH AN ORDER FOR 125 NATURAL GAS BUSES AND ALSO WE UNDERSTAND THAT HAMILTON STREET RAILWAY

WILL SHORTLY ANNOUNCE ANOTHER PHASE IN THEIR NATURAL GAS BUS PROGRAM WHICH WILL INVOLVE MORE BUSES, A FAST FUELLING FACILITY AND DETAILED MONITORING.

THE DIESEL ENGINE NATURAL GAS TECHNOLOGY HAS A GROUND-SWELL OF INTEREST AND ACTIVITY. IT MUST BECOME A TIDAL WAVE, A TSUNAMI. GOVERNMENT AND INDUSTRY MUST PLACE MUCH MORE EMPHASIS ON RESEARCH AND DEVELOPMENT. NOW IS THE TIME FOR ACTION! GET INVOLVED -

THANK YOU!

SECTION

XXVI

DEMONSTRATION OF LNG/DIESEL MINING TRUCK OPERATION

Presented at
Industrial Gas Technology Conference and Exhibition, Vancouver, B.C.
13 - 14 April, 1988

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Inland Natural Gas Co. Ltd.

ABOUT THE SPEAKER:

Bruce Vernon is Product Manager, Transportation Fuel for Inland Natural Gas and is responsible for the Company's activities in the application of natural gas as vehicle fuel. Prior to joining Inland in 1981, Bruce held various engineering positions at BP Oil and Trans Canada Pipelines. He holds a Bachelor's degree in Applied Science (Mechanical Engineering) and a Master's degree in Business Administration.

Inland Natural Gas Co. Ltd. is an investor-owned company which together with its subsidiaries transmits and distributes natural gas to 135,000 residential, commercial and industrial customers in 70 communities located in the interior of British Columbia. Through Trans Mountain Pipeline Company Ltd., Inland participates in the transportation of petroleum and refined products from northern Alberta to the Lower Mainland. In addition, through a subsidiary of both Inland and Trans Mountain the company is engaged in oil and natural gas exploration and development in Western Canada and the United States.

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EXECUTIVE SUMMARY

This paper describes a demonstration of the use of Liquified Natural Gas (LNG) to power large horsepower diesel mining trucks. The demonstration which was initiated in early 1986 is scheduled to be completed in mid 1988. It includes the onsite liquefaction of natural gas from a small skid-mounted liquefier and the conversion of selected mine vehicles to dual fuel natural gas/diesel operation. The project, valued at some $640,000, was funded by Energy, Mines and Resources Canada, Crows Nest Resources Ltd. (the operator of the Line Creek Mine near Sparwood, B.C.), Columbia Natural Gas Ltd. (a subsidiary of Inland Natural Gas Co. Ltd.) and Cryogas Engineering Ltd.

There have been two engine types converted as part of this project: a Cummins NTC-350 and a KTA-1150 that were subsequently installed in a utility vehicle and a coal haul truck, respectively. The utility truck was found during the course of the demonstration to have been an inappropriate selection because of its seasonal use. The coal haul truck, however, has logged some 240 hours on dual fuel operation in revenue service to date and has gone a long way toward satisfying the prime performance criteria of the project - that vehicles converted to natural gas combustion must operate as satisfactorily and as reliably on natural gas as they do on diesel fuel. A number of unique engineering problems arose as the project proceeded which one by one, required identification and resolution.

Overall, operator reaction appears positive: power in the dual fuel mode is equal to that developed in the diesel-only mode and the engine runs more quietly when on natural gas/diesel.

Acknowledgement

Inland Natural Gas is pleased to be an active participant in this project. But it must be recognized that the bulk of the effort in bringing this demonstration to fruition has been expended by Cryogas and Crows Nest Personnel with able assistance from many others including Energy Mines and Resources, Cummins and B.C. Research. This paper is meant to provide only the perspective of the gas distribution utility and therefore does not necessarily represent the views of the other participants.

Background

LNG Vehicle operating experience, especially with diesel-powered units, is very limited. The Atlanta Gas Light Company in the early 1980's operated a utility fleet of some 100 dual fuel LNG/gasoline vehicles and Northwest Natural Gas opened an LNG refueling station on a demonstration basis in Portland, Oregon for a number of their light gasoline-powered vehicles in January, 1984. Other companies such as Shell Research in the U.K. and Beech Aircraft have experimented with the use of LNG in spark ignition engines but application of LNG to compression ignition (diesel) engines has been restricted to stationary and marine engines running for long periods of time under stable operating conditions.

Recent work utilizing compressed natural gas for vehicles (NGV) for diesel engines in a dual fuel application including a number of successful demonstrations (for example, a Ministry of Transportation and Highways dump truck now operating in the Kamloops area) suggests that powering large fuel-consuming vehicles in an off-road application with LNG (with its superior storage density over compressed gas) might now be practical. Within British Columbia the forestry and mining sectors are significant consumers of diesel fuel and might be interested in the prospect of fuel cost savings. The proponent of this project, Cryogas Engineering, approached Crows Nest Resources, who operate the Line Creek Mine near Sparwood, B.C., with an invitation to participate. The open pit coal mining operation employs about forty large diesel powered trucks for coal haul, waste haul and utility services. These together with a number of diesel powered shovels, tractors and graders consume about 14 million litres of diesel fuel per year. The prospect of even a small marginal net improvement in the cost of fuel represents a major opportunity particularly in the coal mining industry which has come under increasing pressure to reduce operating costs.

Project Objectives

The participants in the project forming the management group are as follows:

Cryogas Engineering Ltd. - project design and execution
Crows Nest Resources Ltd.
Energy, Mines and Resources Canada
Columbia Natural Gas Ltd. (sub. of Inland Natural Gas Co. Ltd.)

Additional technical support is provided by:

Cummins British Columbia
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With funding provided by the management group (some $640,000 in total), the purpose of the project is to demonstrate to the mining industry in Canada and to other enterprises which use high horsepower diesel trucks the technical and operating feasibility and the economic benefits of converting such trucks to liquefied natural gas.

The specific objectives include:

1) The installation and operation of a small scale liquefier (owned by Cryogas Engineering) and an LNG storage refueling station at the Line Creek Mine.
2) The conversion and certification of the Cummins NTC-350 and

KTA- 1150 engines to natural gas/diesel operation .

3) The operation and monitoring of the performance of the LNG converted trucks and liquefier.

Performance targets for the project include:

1) LNG Vehicles must operate as satisfactorily and as reliably on LNG as they do on diesel fuel.
2) LNG use must not adversely affect the normal industrial operation.
3) LNG use must enhance the site operator's flexibility in regard to fuel source selection.
4) LNG must result in quickly realizable savings to the site operator.

LNG Production

At normal temperatures and pressures natural gas is lighter than air. When cooled it increases in density until, like most gases, it eventually condenses into a liquid, in this case, at about -160°C. At this temperature it occupies about 1/600th of its original volume (see Figure 1). If heat is added it expands. If released to atmosphere at that point, the gas is cold and is heavier than air but as heat is drawn from the environment it expands and eventually becomes lighter than air.

The process of producing LNG is accomplished by a successive series of cooling steps. After first being cleaned to remove any traces of carbon dioxide and water vapour in a molecular sieve, the gas is compressed (to about 2300 psig in this case) then cooled in a number of stages using a combination of refrigerants and expansions of the gas itself. The Cryogas demonstration unit accepts pipeline quality gas at 25 psig and uses freon as the refrigerant (see Figure 2). About 50,000 standard cubic feet of natural gas per day is consumed a portion of which is utilized by the natural gas powered engine mounted in the unit to provide compression and to power the auxiliaries. Cryogas notes this unit would produce the energy equivalent of about 1300 litres of diesel per day and is a fraction of the size that a commercial scale unit would be. Larger units are found in utility peak shaving plants, which since the late 1960's, have been operating at Vancouver, Sudbury and Montreal to liquify natural gas from the pipeline during periods of low demand and to regasify it as demand warrants in the winter.

The composition of LNG varies over a considerable range and depends on the origin of the natural gas feed, the degree of processing prior to liquefaction, the process used and the degree of "aging" of the LNG in the storage tank. If the liquid is held for a considerable time and the boil off is vented the nitrogen and lighter hydrocarbons will decrease in concentration and the heavier components will increase in proportion. B.C. Hydro reported difficulty three years ago with detonation in a diesel engine utilizing LNG because over time the percentage of heavier hydrocarbons had increased in the storage vessel.

As with any other energy source, LNG must be treated with care. However, because it is a cryogenic liquid only low temperature compatible materials can be used in its handling such as stainless steel or aluminum. Storage containers must be heavily insulated but because no insulation is perfect, a small amount of LNG will continuously vaporize. Since LNG is stored at low pressures (generally less than 75 psig), provisions are made for relieving the storage vessels to avoid a rise in internal pressure. Although LNG by itself will not burn, the natural gas emanating from it is flammable within certain limits (5-15% natural gas by volume).

Particular attention has been paid in this project to LNG containment which includes a 13000 litre storage vessel (see Figure 3) and six seventy-five litre cylinders on the coal haul truck (see Figure 4). Transfer of LNG to the truck was accomplished through a stainless steel coupling (see Figure 5).

Diesel Conversion

In principle, natural gas may be introduced into a diesel engine either by induction into the intake manifold or directly into the cylinder. This project has utilized direct injection such that at full power the pilot diesel fuel may account for only 20% of total fuel. As power decreases, the pilot charge makes up an increasing percentage of total fuel until, at idle, the engine is operating exclusively on diesel. This variable pilot to gaseous fuel ratio requires a sophisticated control system for optimum fuel utilization. Cryogas has developed a microprocessor - based system to accomplish this task. Fuel switching from diesel to dual fuel mode and back again is accomplished electrically from the driver's cab.

Two vehicles have been converted at Line Creek - a utility water truck equipped with a Cummins NTC-350 (300 HP nominal) and a coal haul truck equipped with a Cummins KTA-1150 (500 HP nominal). The utility truck was found during the course of the project to have been an inappropriate choice because of infrequent, seasonal use and the conversion equipment was removed. The coal haul truck currently remains in operation. Prior to conversion of the vehicles, each engine type was dynamometer-tested at the British Columbia Institute of Technology under the supervision of Cryogas and B.C. Research. Difficulties were later encountered with the KTA aftercooler performance on natural gas.

Interim Operating Results

1. To date some 80 hours of operation have been logged on natural gas operation, equivalent to about 240 hours of shift operation since natural gas is consumed only on the uphill portion of the coal haul truck's route (the truck is loaded with coal on the descent and the engine - running on diesel - is used in a retarding mode).
2. Driver reaction appears positive: the response and power is equal to that in the diesel-only mode and the engine noise is noticeably decreased. There has been no difficulty switching fuels back and forth and the conversion has not affected the diesel-only operating mode.
3. Mine operations management reaction appears generally favourable although scheduling the converted coal haul truck for repairs has been disruptive. The mine operates seventeen such haul trucks three of which are normally down for repairs at any one time and one is used for refuse hauling. The 13 remaining allow for no additional spares and the maintenance schedule for all units is very tight.
4. No particular safety concerns have been expressed; however, all refueling has been accomplished by Cryogas personnel to date.
5. Early difficulties with freeze up of the refueling probe to its receptacle have necessitated a design change to the probe which was incorporated by Cryogas.
6. Operation of the liquefier was performed by Cryogas - mine personnel have not been involved in LNG production to date.
7. The integrity of the insulation of the storage containers has been an ongoing source of frustration during the course of the project. Several of the cylinders were replaced with reconditioned units but excessive boil off has caused the truck not to be used during night

shifts. One or two larger new cylinders may have been a more appropriate choice for onboard vehicle storage. Through most of March only four of the six cylinders were actually used in order to minimize the venting of gas and consequently to avoid the degradation of the LNG left in the cylinders.

8. The electronic control unit has suffered a number of faults resulting in the entire loss of programming on one occasion (related to spurious electromagnetic interference) and incorrect signal processing on another (an intermittent problem that has not been fully identified at the time of writing this paper).
9. Total refueling time appears to take about 20 minutes. Of that 10 minutes are required for the actual transfer of LNG from storage to the vehicles four operating cylinders.
10. A mass flow meter has now been installed on the refueling facility to record the amount of LNG actually transferred. This will allow the participants to draw some conclusions about fuel economy and potential fuel savings, a necessary step before consideration can be given to expanding the scope of the project.

Conclusion

This unique project continues to yield important data concerning the application of natural gas to severe service, large diesel engines operating in a dual fuel configuration. It also is providing specific field experience on the production and handling of LNG in an industrial setting. At the completion of the project the results will be summarized and a final report made available.

The specific objectives of the project have been met: the liquefier was installed at the mine site along with a refueling facility; the Cummins NTC-350 and KTA-1150 engines were successfully converted to natural gas/diesel operation; monitoring of the performance of both the coal haul truck and the liquefier have been underway for several months.

The project must now address the performance targets set at the inception of the project. It has gone a long way toward satisfying one of the prime criteria - that vehicles converted to natural gas combustion must operate as satisfactorily and as reliably on natural gas as they do on diesel fuel. It has been clearly shown that a converted coal haul truck can meet the demanding requirement of a full power return trip up the road to the minesite and it has done so to the satisfaction of the vehicle's operator. What remains to be demonstrated is that the use of LNG does not adversely affect the normal industrial operation - while refueling times are probably acceptable the maintenance frequency may not be. And it must be shown that LNG offers both fuel source flexibility and realizable cost savings.

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North Bay, Ontario
P1B 8K7

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IMW Compressors
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Chilliwack, B. C.
V2P 1B5

Mr. Fred Feistmann
IMW Compressors
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Chilliwack, B. C.
V2P 1B5

Mr. David Stevenson
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V6P 6E1

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International Fuel Cells Corp.
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IPSCO Inc.
P.O. Box 1670
Regina, SK
S4P 3C7

Mr. Ron Duncan
Island Paper Mills Ltd.
Box 2170
New Westminster, B. C.
V3L 5A5

Mr. Don Niquidet
Jacobson Bros. Forest Products Ltd.
P.O. Box 4710
Williams Lake, B. C.
V2G 2V7

Mr. Rick Zilkie
Johnson Controls
3061 Grandview Hwy.
Vancouver, B. C.
V5M 2E4

Mr. Paul Levert
John Zink Canada Ltd.
#3 - 3571 Viking Way
Richmond, B. C.
V6V 1W1

Mr. Don Geesen
Kalium Chemicals

Mr. Bryan Drew
Kilborn Engineering
12921 99th Avenue
Surrey, B. C.
V3T 1E6

Mr. Leo Peterson
Labatt Breweries of B. C.
210 Brunette Avenue
New Westminster, B. C.
V3L 4Z2

Mr. Dave Taylor
Labatt Breweries of B. C.
210 Brunette Avenue
New Westminster, B. C.
V3L 4Z2

Mr. Wayne Brooks
Liquor Distribution Branch
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V5M 1Z6

Mr. J. G. Sanderson
MacMillan Bloedel Ltd.
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V6E 3R9

Mr. Jim Halldorson
Matsqui Institution
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Abbotsford, B. C.
V2S 4P3

Mr. Ralph Huston
Matsqui Institution (CSC)
P.O. Box 2500
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V2S 4P3

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V4K 3G1

Mr. Cye White
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98055

6 May 1988
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N.W. Natural Gas Co.
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97209

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Northwestern Utilities Ltd.
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Edmonton, Alta.
T5J 2V6

Mr. Peter M. Lowe
Commercial-Industrial Marketing
Northwestern Utilities Ltd.
10035 105 Street
Edmonton, Alta.
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Mr. Fred Marlett
Northwestern Utilities Ltd.
10035 105 Street
Edmonton, Alta.
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Mr. Allen J. Pirot
Northwestern Utilities Ltd.
10035 105 Street
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Persica Enterprises Inc.
4479 Bradner Road
Aldergrove, B. C.
V0X 1A0

Mr. Dieter Kirste
Pine Mechanical Ltd.
#201 - 6958 Palm Avenue
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V5J 4M3

Mr. Robert Goth
Plainsman Mfg. Ltd.
5928 99th Street
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T6E 3N9

Mr. Steve Kwai
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1166 Alberni Street
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V6E 3W5

Mr. Norman Ayre
Reichhold Ltd.
50 Douglas Street
Port Moody, B. C.
V3H 3L9

Mr. J. Isaak
Ritchie-Smith Feeds Inc.
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V2S 4N9

Mr. H. L. Roeters
S & R Sawmills Ltd.
18887 98A Avenue
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V3T 4W2

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700 - 1190 Hornby Street
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V6Z 2H6

Mr. George Sigal
Sandwell Swan Wooster Inc.
700 - 1190 Hornby Street
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K4.28

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6 May 1988
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